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THE

GAS-ENGINE HANDBOOK,

A MANUAL
OF USEFUL INFORMATION
FOR THE

DESIGNER AND THE ENGINEER.

BY

E. W. ROBERTS, M. E.,

Author of The I. C. S. Textbooks on Gas, Gasoline and Oil Engines.

SECOND EDITION. SECOND THOUSAND.

18

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NOTE.

It is the earnest desire of the publishers and of the author, that this work shall be a standard of reference for those interested in gas enginery. They will, therefore, be pleased to have their attention called to any errors of commission or omission in this book.

THE PUBLISHERS.

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PREFACE.

It was during the preparation of a series of textbooks on the gas engine for the International Correspondence Schools, that the author was most forcibly impressed with the dearth of matter upon American practice in this motive power. It is a recognized fact that designers on the other side of the Atlantic do not follow methods that meet with the approval of engineers in the United States, yet the only truly valuable works on gas-engine design that have made their appearance in the English language are by English authors. Unhappily, the average gas-engine manufacturer in this country, guards any information he may possess with the jealousy that is scarcely to be explained on ordinary grounds. While there are a number of good works on steam-engine design, the gas engine has been surprisingly neglected. Although the author does not, in the present work, hope to cover the entire subject gas-engine design he has endeavored

dimensions of some of the most su
American-built engines, and hav
used by the author in making up
designs.

It has been the endeavor, to plac
the smallest possible compass a nu
useful rules and hints, that may be
not only to the designer, but als
engineer who has the care of a gas
The chapter on testing has bee
more attention than might perha
been thought necessary in a book
size, but it also covers many points
ing the calculation of horsepow
other items purposely omitted f
other portions of the book. This
of the work was founded upon the
employed at The Cornell Universit

referred to. A part of the data in the table of heat values is from a similar table in the work of Mr. Gardner D. Hiscox. The mechanical tables are from various pocket-books and from the works already referred to.

Much that the writer would have desired to include in the present work, it has been necessary to omit for lack of space. He has, therefore, given only such matter as he has judged most useful, and which he has found most frequently the subject of inquiries in the question and answer columns of the technical papers.

E. W. ROBERTS.

Cincinnati, March, 1900.

SECOND EDITION, OCTOBER, 1900.

In this edition all typographical and other errors that have been found have been corrected. It has also been found advisable to enlarge the book by the addition of an appendix, so as to include some notes on two-cycle engines, and especially upon their design. For the data on two-cycle engines in the appendix, and for nearly all of the notes on this subject, the author is indebted to Mr. Harry J. Perkins. The author also wishes to extend his thanks to the many readers of the book who have helped him with timely suggestions, a number of which appear in the notes in the appendix.

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CHAPTER I.

INTRODUCTORY.

The Gas Engine, properly so called, is a type of the internal-combustion engine. Its first conception seems to have been by one Abbe Hautefueille, who proposed a motor driven by a gunpowder as early as 1678. From that time until 1862 many propositions were made and some partially successful engines were built by various experimenters. In the latter year, there were already in existence the gas engines of Lenoir and Hugon, but very wasteful of fuel. In 1862 M. Beau de Rochas, a French engineer, made the following propositions, which he embodied in a patent of that date. They were as follows:

I. The largest cylinder capacity with the smallest circumferential surface.

II. Maximum speed of piston.

III. Greatest possible expansion.

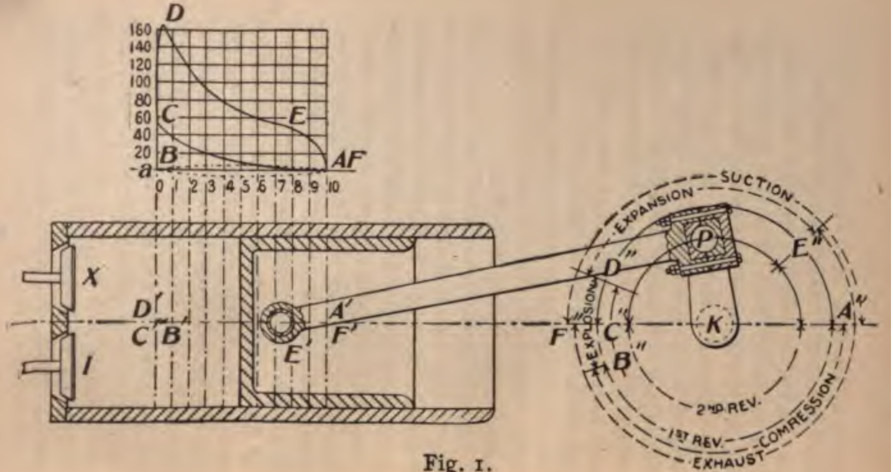
IV. Highest pressure at the beginning of *the stroke.*

These, he averred, should be the aim of every gas-engine designer to carry out in his engine. With the exception of proposition II, every engine, since the successful advent of the Otto engine, has been designed with these principles in view. Owing to certain practical limitations, high piston speeds have not been found advisable.

It was M. Beau de Rochas who first patented what is popularly known as the "Otto Cycle." This cycle requires four strokes of the piston for its completion. The first stroke draws in the charge of gas and air, the second stroke compresses the charge, on the third stroke ignition takes place at the dead point, followed by explosion of the gaseous mixture and the expansion of the products of combustion, and on the fourth stroke the burnt gases are expelled from the cylinder. This series of operations is that employed by nearly all gas engines in use at the present time.

This cycle was not put into a practical use in a working engine until, in the year 1876 Dr. N. A. Otto brought out the first "Otto Cycle" engine, producing a motor which so far surpassed in economy all engines that superseded it as to drive its competitors from the field. This cycle of operation has, because of its being first put into practica shape by Dr. Otto, been improperly termed the "Otto Cycle," although the credit of its first inception belongs, without question, to M. Beau de Rochas. It is also known as a four-cycle engine, although this term, to be strictly correct, should be four-part cycle, because it consists of the four distinct operations just described. Modern writers have been placing the credit where it belongs, and this series of operations is now generally termed the Beau de Rochas cycle. The author, however, prefers the same descriptive term, "four-cycle," and as such it will be referred to throughout this work.

The practical operation of the four-cycle engine can be most clearly understood by reference to Fig. 1, in which a diagramatic form of the engine is shown with the indicator diagram given by the engine placed directly above the cylinder. Since it requires two revolutions of the crankshaft K to complete the cycle, the principal points of the cycle are shown on the two circles drawn about the crankshaft as a center. Similar points upon the diagram, the piston travel and the rotation of the crankpin P, are indicated by the same letters. The letters indicating points on the piston travel are primed, and those indicating similar points on the crank circle have a double prime. *Thus*, the point of ignition is indi-


D
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X
I
D′
C
B
E
A′
F′
EXPANSION
SUCTION
P
E″
D″
EXPLOSION
F
C″
B″
K
A″
2ND REV.
1ST REV.
EXHAUST
COMRESSION

Fig. 1.

cated on the diagram by B′ on the line of piston travel, and by B″ on the crank circle. Taking the series of operations in their regular order, they are as follows:

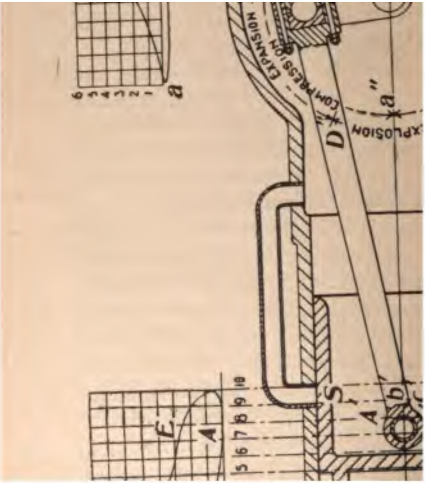
Suction of air from a to A on the outward stroke, compression from A to C on the inward stroke, ignition taking place at B, just before the end of the stroke is reached, thus giving a "lead" to the ignition which should just be sufficient to bring the point of maximum pressure D, at the beginning of the next outward stroke; this lead being necessary because of a lag in the process of ignition, which takes the time required by the piston going from B to C to fully inflame the charge of gas and air. From D to F the piston is passing on a second outward stroke, the exhaust-valve X opening at E, just in time to allow the expansion line D F to reach atmospheric pressure at the end of the second outward, or expansion, stroke. The return stroke takes place from F to a, the piston making a second return stroke, driving the products of combustion through the exhaust-pipe by way of the valve X, completing the cycle.

A modification of the four-cycle engine was brought out by its inventor, Mr. Dugald Clerk, in the year 1880. In this engine, using an auxiliary pump, a power stroke was *obtained* for each revolution of the crankshaft. The charge was admitted to the engine cylinder under pressure and just at the end of the expansion stroke, driving the exhaust gases out through ports in the sides of the cylinder, which were uncovered by the piston just at the end of the stroke. This arrangement eliminated both the suction and the exhaust strokes required in the regular Beau de Rochas cycle. Clerk's engine was the pioneer of what is now known as the two-cycle type; that is, an engine which completes its cycle in one revolution of the crankshaft.

Following the introduction of the Clerk engine came a class of engines which, instead of having an auxiliary cylinder to compress the charge, used for that purpose an enclosed crank-chamber. The inward stroke of the piston draws into the crank-chamber the charge of gas and air through a check-valve, and on the outward stroke the charge is compressed. Just before the piston reaches the end of its stroke its end passes an exhaust-port in the side of the cylinder, and in some engines a port leading to the crank-chamber is passed immediately afterward.

In another type, communication was made by means of a poppet valve. The pioneers of this type were the Nash and the Day *engines.* The former used a mechanically opened poppet valve, and the latter an admission port in the cylinder wall. Quite a number of engines of the Day type are in successful operation at the present time.

The principle of the operation of the Day type of the two-cycle engine is shown in Fig. 2, in a similar manner to that of the four-cycle engine shown in Fig. 1; the same method of notation as regards letters being used in this figure, the only difference being that where the two crank circles are used, the inner one indicates the series of operations taking place in the crank-chamber, and the outer indicates the series of operations taking place in the cylinder. The piston makes an inward stroke from b to a, causing a partial vacuum in the crank-chamber and drawing the explosive mixture through the valve P. On the following outward stroke the mixture in the crank-chamber is compressed to a pressure of about five pounds to the square inch. As the piston uncovers the inlet port S, the pressure within the crank-chamber drives a portion of its contents into the cylinder C, and on the next inward stroke of the piston this charge is compressed, as shown on the diagram from A to C, ignition taking place as in the four-cycle at B, pressure rising to D, the piston making outward stroke, exhausting at E, and the gases reaching atmospheric

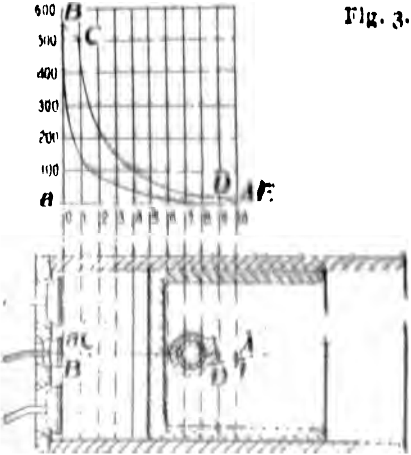

EXPANSION
COMPRESSION
EXPLOSION
D
a"
a
1
2
3
4
5
6
E
A
S
A
b
c
5
6
7
8
9
10

pressure again at A, the exhaust-port being opened by being uncovered by the piston. In the meantime a charge is being drawn into the crank-chamber as before, has been compressed, and the inlet-port S being uncovered immediately after, the contents of the crank-chamber rushing through S is deflected by means of the plate R on the piston to the top of the cylinder, effectually driving the products of combustion through the port M. Returning to the series of operations in the crank-chamber, the port S opens just as the piston reaches the point E, the pressure immediately dropping and reaching the pressure of the atmosphere at about the time port S is closed by the return of the piston. It is unnecessary to repeat the explanation of the points on the diagram taken from the cylinder C, as they are in all respects similar to the four-cycle engine illustrated in the Fig. 1.

The Clerk engine is no longer built. Nearly all two-cycle engines being manufactured at the present time are built after the Day pattern. In the latter part of 1897, a new cycle was brought to the attention of the public. This cycle comprises the series of operations which were the invention of Herr Rudolph Diesel, a German scientist. The Diesel cycle is, like the invention of Beau de Rochas, a four-part

entire cylinder capacity, the pres
end of the stroke being approxi
pounds per square inch. This
pression produces a temperature
equal to that produced by the c
of the fuel, and, in consequenc
mitted at the end of this strok
be ignited as it enters the cyli
fuel, whether it be gas, oil or
bustible material, is forced into
der at a pressure higher than tha
by compression, and just as th
about to start on its outward st
fuel, burning as it enters, keeps
ature of the cylinder contents
produced by compression. Aft
ton has reached a point of

Fig. 3.



engines using the cycle of Beau de Rochas, and that but a very small quantity of the products of combustion remain to mix with the fresh charge. This series of operations may be better understood by reference to the Fig. 3.

The same manner of representation is employed as in Fig. 1. Suction of a charge of pure air takes places from a to A on the outward stroke, the following inward stroke compressing this charge until the point B is reached, when the fuel is admitted while the piston is passing from B to C and is cut off. The temperature of the gases is constant during admission of the fuel, the line from B to C being one of equal temperature known as an isothermal. From C to E, expansion proceeds in a manner similar to that in the original four-cycle engine, the exhaust-valve opening at D and the pressure inside the cylinder falling to atmospheric at E. The range of temperature from C to D is much greater than that from D to F, in Fig. 1, and represents a much higher heat efficiency than that shown on the diagram of the four-cycle engine.

CHAPTER II.

COMPARISON OF THE THREE CYCLES.

The four-cycle engine is that most generally manufactured, and it has the advantage over the two-cycle of being more readily controlled. With a properly designed four-cycle engine, the behavior of the gases within the cylinder is known beforehand. The idle stroke gives the cylinder a chance to cool by radiation, no pumps nor enclosed crank-chambers are necessary, and any waste of fuel may be easily remedied. On the other hand, the four-cycle engine must be built in large sizes when compared to the power demanded of them, the many idle strokes necessitate extremely heavy flywheels and make close regulation most difficult to obtain. Furthermore, the operation of the valves occurs but once during two revolutions of the crankshaft and necessitates some form of reducing motion between the crankshaft and the camshaft which operates the valves.

The *two-cycle* engine is, as a rule, more

in the crank-chamber. The bu
troubles is fortunately confined t
built by designers who have an
knowledge of the type, and who
sign their engines with the prop
tions.

Time should always be given th
gases to fall below the pressur
crank-chamber before the inlet-p
cylinder is uncovered. It is also i
when running a two-cycle engi
member that much more heat
given off through the cylinder wa
a four-cycle engine of the same p
ing the same period, hence the
jacket water must be greater in co

A two-cycle engine will shut d
quickly from lack of water than w

These engines are being better understood as the makers gather knowledge from experience, and the above objections have, in several cases, been entirely overcome. So many and manifest are the advantages of more frequent impulses and less weight, both in the engine itself and in the flywheels, that the adoption of the two-cycle type is being seriously considered by a number of manufacturers. In fact, the two-cycle engine, for large power-units where the fuel employed is blast-furnace gas, has been in use for some time and found to give satisfaction. Instead of having an enclosed crank-chamber, these engines are supplied with a pump which first drives a volume of pure air through the cylinder, clearing it entirely of the products of combustion remaining from the previous charge. The fresh mixture follows the charge of pure air, and none of the fuel is wasted, while the clean cylinder increases both the efficiency and the capacity of the engine.

The Diesel motor is not at present upon the market in sufficient numbers, nor for a length of time, that would make it feasible to draw conclusions of any kind with regard to the future of the engine. The engine is a grand conception theoretically, and the heat efficiency is much higher than that of *any heat engine* in use. The scheme of regulation is similar to that which has proven so successful in high-speed steam engines, and it has been shown by actual tests that, in practice, the engine surpasses any other in economy of fuel, and, besides, it is capable of being run with such inexpensive fuels as the low-grade petroleums classed as fuel oils and selling in quantities for from two to three cents per gallon. The question therefore remains: will the great economy of the motor justify the added expense for first cost of the engine? Again, there is the question of repairs and the durability of the engine under the high pressures which it is necessary to sustain in the cylinder. All this remains to be seen.

CHAPTER III.

GAS-ENGINE FUELS.

For driving a gas engine, the fuels available are all of those which are in use for other purposes. Many of these fuels need to be transformed into gas before they are available for the gas engine, but many of the liquid and all of the gaseous fuels may be employed directly in the cylinder without the necessity of an intermediate process. There is no fuel not in a gaseous state already that may not be transformed into gas by one of the many processes known to science. The garbage gathered from the streets of our large cities, the waste fat from the slaughter-house, the fat abstracted from wool, as well as all of the animal, vegetable and mineral oils, have been at one time or another transformed into gas which could be used to drive a gas engine.

The amount of power derived from a certain *quantity of* fuel is always greater by a large amount when the fuel is first made into gas and then used to drive a gas engine, than when the fuel is consumed in the firebox of a steam boiler and the power derived from a steam engine. Even with coal as a fuel, if the coal be made into gas and the gas used in a gas engine, it is found that where the very best steam engines give a horsepower for one hour on 1½ lbs. of coal, the same amount of fuel has given in a gas engine 1.8 horsepower for one hour.

The gas-producer which is required for the use of coal in connection with the gas engine, is a much simpler device than the steam boiler, and requires less attention than the latter. There is very little danger of explosion with a gas-producer and gas-engine power plant, and there is usually a supply of gas on hand in the gas-reservoir sufficient to permit the engines to be started at a moment's notice without the necessity of waiting, as when getting up steam for a steam engine.

Fuels which are most difficult to burn under a boiler, may be turned into gas in a gas-producer. Even gases which are difficult to ignite may be used in a gas engine. *For instance*, the gas produced during the *process of* extracting iron from ore in th *blast furnace* is at times so poor in quali *that it may not* be used under a boiler, wh when compressed in a gas-engine cylinder it is found to ignite readily.

The value of fuel for use in a gas engine, is determined in a great measure by the number of heat units produced when a quantity of the fuel is burned. There are, however, several other considerations, such as the pressure derived from the fuel and the percent that may be burned in the gas-engine cylinder. In the process of making gas from fuels a percentage of the heat is employed in liberating the gas, and this, of course, can not be given account of in the gas engine.

The heat values of the various fuels are given in Table 1. It will be seen that the highest heat value is found in natural gas. This is due to the large quantity of marsh gas contained in natural gas. The value of natural gas as a fuel varies, even when the gas is taken from wells very close to one another. It may be said, however, that the natural gas obtained from the wells in western Pennsylvania exceeds in heat value that of the gas in the Ohio fields from 10% to 15%.

Gasoline, in spite of the low heat value per cubic foot of vapor shown in the table, will give, for the same size engine, a power equal to that derived from the best natural gas, *and the usual* custom is to credit gaso-

hence a larger quantity of g
into the cylinder of the
stroke.

FUEL.	B. T. U. per lb.	B. T. U. per cu. ft.
Hydrogen @ 32° F. . . .	62,030	348
Carbon	14,500	
Carbon monoxide (CO) .	4,396	539
Penn. heavy crude oil. . .	20,736	
Caucasian heavy crude oil,	20,138	
Caucasian light crude oil,	22,027	
Petroleum refuse . . .	19,832	
Anthracite gas.	2,248	
Bituminous gas	3,484	
28-candlepower illum. gas,	. . . .	950
19- " " "	. . . .	800
15- " " "	. . . .	620
New York city water-gas*,	. . . .	710.5 Ave
London coal gas. . . .	. . . .	668
Benzine C_6H_6	18,448	
Gasoline and its vapor . .	21,900	690
Ethylene C_2H_4	21,430	1,677
Marsh gas (Methane) CH_4	23,594	1,051
Nat. gas, Leechburg, Pa.,	. . . .	1,051
Nat. gas, Pittsburg, Pa . .	. . . .	892
Acetylene C_2H_2	21,492	868
Semi-water gas	. . . .	185
Producer gas	. . . .	150

NOTE.—The values shown in the above table are given on what is deemed good authority, but they will not be found to agree with all similar tables.

---

* Carbureted gas at 60° F. and at 30″ water pressure.

CHAPTER IV.

STARTING AND STOPPING.

Starting a gas engine is a simple operation if a few easily-remembered rules are borne in mind. These rules are briefly as follows:

A gas engine will not start itself, like a steam engine, and must receive sufficient turning power from an outside source to enable it to take up its cycle of operations.

The mixture of fuel and air must not be too rich in fuel nor too poor in fuel, for in either case an explosive impulse can not be obtained. More frequently, trouble occurs from the mixture being too rich in fuel.

On large engines the compression should be relieved and the ignition given a negative lead, especially when starting by hand.

Never attempt to start a gas engine when it is connected to a load, unless it has a powerful starter.

The ignition apparatus must be in good.

working order, and this must be seen to before attempting to start the engine.

To start a gas engine, it is always a good idea to have a regular order of procedure and to stick to this order so as to avoid the chance of confusion and of omitting something. A good way to proceed is as follows:

First, oil the engine thoroughly, filling every oil-cup, no matter if one or more is already nearly full.

See that the ignition appliance is in good order. If it is a tube igniter, bring the tube to the proper temperature (usually a cherry red) and see that the flame is at the right height to keep the tube at the required heat. If an electric igniter is used, unfasten the wire from the insulated electrode and brush it on some part of the engine to see that it gives a good flash.

If there is a sight plug, it would be well to try the igniter while observing its behavior through the hole, or it can be tried by first pressing the points together by means of the igniter mechanism and seeing if there is a circuit by drawing the wire already removed over the end of the insulated electrode, then letting go of the mechanism and determining in the same manner as before if the circuit is broken.

If these conditions are fulfilled, and when the *engine is turned* over it is found to close the circuit on the igniter-points, the igniter is in working order.

The igniter should be examined periodically and the points cleaned of any accumulation of soot or other foreign matter which may occur. The frequency of this examination depends entirely upon the fuel used, and to some extent upon the class of cylinder oil employed.

After determining that the igniter is in working order, such oil-cups as can not be conveniently reached when the engine is running should be turned on.

Next set the starting-cam to start, or open the relief-cock, as the case may be, and if the engine is fitted with a changeable igniter, set this also to the starting position, or so that it will fire after the crank has passed the center.

If the engine is to be started by hand, set the gas-valve open to about one third the opening used when running; usually at a point on the gas-valve dial marked "Start." Then turn the engine over in the running direction until it takes up the cycle, *i. e.*, until an explosion is obtained.

Next take hold of the gas-valve handle *immediately*, and open it gradually to the *running position* as the engine increases in speed. *Don't be too precipitate about opening the gas-valve*, as the engine may get too rich a mixture and, failing to ignite, slow down and stop. If it shows signs of stopping, close the valve a little at a time until it starts receiving impulses again. While opening the gas-valve with one hand, the other may be occupied in closing the relief-cock, or in throwing the starting-levers to the running position.

When the engine has reached its full speed, the load may be thrown on, and then the remaining oil-cups must be opened and the water turned on to the water-jacket.

After the engine has been running a short time, and if the load is very nearly a constant one, the water should be turned on or off until the exit water is of a temperature that can just be borne comfortably by the hand.

If the engine is fitted with a starter, the gas-valve should not be turned on until the engine has made one or two revolutions. The methods of using the starter vary with the several types, and their use will be explained in a chapter on that subject.

Simply to stop a gas engine, it is necessary to do no more than turn off the fuel. The order for turning off the oil-cups, etc., should be about the reverse of starting.

First turn off the gas, and if it is desired to save time in stopping, brake the flywheel with a *plank*. Then turn off the jacket

prevent the accumulation of se
the jacket, and it takes but little
habit of draining the cylinder m
engine at some time when you " d
it was going to freeze."

CHAPTER V.

CARE OF AN ENGINE.

The proper care of a gas engine should be the pride of every engineer who runs one. In every shop or factory where a gas engine is in use, it should be left in the care of one man and that man should be responsible for the condition of the engine at all times. "Everybody's dog is nobody's dog," and so it is with an engine which is left in the care of any man who happens to be about at the time it needs attention. Above all things keep the engine clean, well oiled and with a plentiful supply of water in the tank, when a tank is used, or with sufficient flow of water when connected to a pressure system.

If the engine is a new one when first received, it is well to determine the exact point of the stroke where the ignition takes place. This can be done very easily, with an electric igniter, by turning the engine over until the igniter snaps for breaking the *circuit, and* marking the piston so as to have a point of reference for future settings of the igniter. All igniter mechanisms wear more or less, and the mark on the piston will avoid the necessity of taking an indicator diagram to set the igniter.

If the piston is so situated as to make it inconvenient to make a mark upon it, the mark may be placed upon the flywheel. To mark the flywheel for this purpose, proceed in one of the following ways: Turn the engine over slowly until the igniter just snaps. If the engine has been turned too far it will be necessary to make a complete revolution again in order to be certain that all lost motion is taken up. After the flywheel is in the right position, drop a plumb-line past the center of the crankshaft and mark on the rim of the flywheel opposite where the line passes it, or level up a straight-edge so that it passes the center of the crankshaft and mark the flywheel as with the plumb-line. It will be more convenient even than the above methods, if the flywheel passes close to some part of the engine-frame or bed, to mark both the wheel and some point on a stationary part of the engine.

A good grade of machine oil should be used on the engine bearings, but in the cylinder there should always be used an oil that is made expressly for this purpose. *Never use the heavy* cylinder oils that are sold for steam-engine cylinders or you will soon find that the passages are becoming choked with carbon. The proper oil for use in a gas-engine cylinder is a thin lubricant which will not carbonize under the high temperatures present in the gas-engine cylinder. Don't think that because steam-engine cylinder oil costs four times as much as gas-engine cylinder oil it is four times as good. The direct opposite is true. For an engine with an inclosed crank-case another special oil is required and is known to the trade as "crank-case oil." Ordinary oils will churn into lumps in the presence of water and soon become practically useless as a lubricant.

It is not a bad practice to make an occasional examination of the exhaust-passages in order to determine whether they have an accumulation of carbon in the form of soot. With some gases this deposit does not amount to a great deal, while with others the deposit is such as to in time cause sufficient back pressure to materially decrease the power of the engine. Should the water used for cooling purposes contain substances that are likely to cause a deposit of sediment in the water-jacket, the jacket should be cleaned occasionally by means of a long iron rod with a hook on the end, *similar to a* poker.

Examine all valves occasionally, paying particular attention to the exhaust-valve to see if it is cutting. If they show the least sign of leaking they should be attended to at once and ground to a good fit to the seat with flour emery. Never neglect a leaky valve, for, once started, the leak will increase rapidly in size: Examine the springs frequently to see that they pull the valves securely to their seats. This can be determined by taking hold of the valve-stem and pulling the valve from its seat. The resistance to the pull should be a stiff one, even at the start, for all valves excepting a suction inlet. In the case of an inlet-valve operated by the suction of the piston, it should respond promptly to the vacuum in the cylinder, a very small movement of crank past the center causing it to open. It should be carefully adjusted so that it will seat firmly and yet respond promptly to the vacuum. The gas-valve should also receive its share of attention.

With engines using electric ignition, the care of the igniter mechanism is one of the most important points. Never neglect the igniter. Much of the prejudice against the electric igniter is the result of improper care of the ignition apparatus. In case a primary cell is used for the battery, a full *set of renewals* should be kept on hand at all times and the battery should be renewed before it is too weak.

In case an ignition-tube is employed, it pays at all times to get the best that can be obtained. Nickel alloy is much the best material to employ for this purpose, as a tube made from the alloy will last, with ordinary care, from six to eighteen months, according to the work the engine has to perform. Wrought-iron tubes have to be replaced anywhere from every two days or so to every two weeks. The shut-down necessary while replacing a tube is annoying, to say the least, and it may cause expensive delays. A new tube should be kept in stock at all times, and when tubes that last but a short time are used, it would be well to keep several on hand.

Do not keep the tube too hot at any time, as a high temperature reduces its strength and makes it wear out so much sooner than it otherwise would. The proper temperature to keep the tube is at a bright cherry, not a white heat. The temperature of the tube for ignition, will vary somewhat with gas of different qualities and the engineer should determine for himself the lowest temperature at which the tube will work successfully.

If at any time the engineer has forgotten to turn on the jacket water and the engine begins to throw out volumes of smoke through the open end of the cylinder, turn on the jacket water, but do so with great caution, as, if the cylinder walls are cooled too rapidly, the cylinder may contract before the piston has time to do so, and it may bind the piston so as to materially damage the engine. The cylinder oil-cup should also be opened wide at the same time, to give the cylinder an excess of oil and ward off the possibility of its binding.

Feel of the bearings occasionally, and keep watch of the sight feed of the oil-cups, so that no injury may be done to the engine by cutting of the journals. Don't attempt to cool an overheated bearing with ice unless you thoroughly understand what you are about. You may get an effect the opposite of that desired, and find that the sudden cooling of the outside of the bearing has caused it to grip the shaft like a vise. If the ice can be applied to the shaft itself so as to cause the shaft to contract before the surrounding metal, all trouble may be averted and the desired effect obtained.

Examine the governor occasionally to see that it is working freely and is not clogged by dirt or any foreign matter. The regularity of motion is dependent as much upon the sensitiveness of the governing device as *upon the flywheel.* If the governor is a hit-

and-miss, see that the gas-valve is open just far enough to cause an explosion to follow immediately upon the action of the governor. Nearly all governors of this type use a pick-blade, and if the gas is not turned on sufficiently, the blade will strike or let go — according to its method of operation — some time before the engine will receive an impulse, and the consequence is a slowing down of the engine until the mixture becomes rich enough to produce an explosion.

If there is a pet cock on the engine having communication with the compression space, the proper state of the mixture may be determined by opening the pet cock just at the time of ignition and observing the color of the flame. If the flame is an extremely pale blue, the mixture has too little gas and the gas-valve should be opened a little wider. If, on the other hand, the flame is tinged with red, orange or yellow the mixture is too rich. The proper color of the flame is a deep blue approaching a violet. Do not depend for this test upon a pet cock situated close to the gas-valve opening, as the mixture at this point of the compression space is usually much richer in gas than at the center.

At intervals of six months, if the engine is in *constant* use, remove the piston and is especially true of the exhaust-va

CHAPTER VI.

GAS-ENGINE TROUBLES.

While it is not possible to anticipate every trouble that may occur in the running of a gas engine, a general outline of such troubles as are of frequent occurrence, and the remedies that apply in each case, will give a clew to the solution of those problems not considered in this chapter.

Failure to Start.—See first that the fuel-valve is not open too wide, nor that it has not been open a small amount for a length of time that would allow an excess of gas or gasoline to leak into the cylinder. Then examine the igniter. If a tube, see that it is not too cold. If an electric igniter, look it over thoroughly as directed in the chapter on Starting. See if the gas supply is interrupted in any way. It may not have been turned on. In cold weather, a gasoline engine may fail to start because the gasoline does not vaporize. In this case, warm the air-inlet pipe, or fill the water-jacket with hot water. In very severe weather it is a good plan to do both.

Engine Starts, but with Weak Explosion.—The igniter may be set too late, and it should be adjusted to ignite just before the engine reaches the rear dead center on the compression stroke. The fuel-valve may not be open wide enough. The engine may lose compression from various causes. One or both of the valves may leak or may not seat itself with sufficient force. The starting lever may be still in the starting position. The relief-cock may be open. The piston may leak, but this is not often sufficient to materially affect the power of the engine. See that the valve springs are not too weak.

Explosions in the Exhaust Passages.—The exhaust-valve may leak or may not seat properly. The ignition-tube is too cold, or the electric spark is weak. The fuel-valve may be open too wide, so that, occasionally, the mixture in the cylinder is too rich to take fire. The primary cause of this trouble, is always the occurrence of unburned fuel in the exhaust passages.

Engine Slows Down and Finally Stops.—This may be due to overheating of the piston or the cylinder, because of an insufficient supply of water, or insufficient oil. One or more of the bearings may be overheated, notably the crankpin or the crankshaft bearings. The engine may be overloaded. See also the comment under *Weak Explosion* above. For the treatment of an overheated cylinder or bearing, read the chapter on the care of a gas engine. See paragraph on "Back Firing."

Explosions Cease. — See paragraph on *Failure to Start.*

Premature Explosions, "Back Firing." — This trouble is of most frequent occurrence with fuel of a low ignition temperature, such as gases rich in hydrogen, and gasoline. If the ignition apparatus is properly adjusted, the source of the trouble may be traced to an overheated cylinder and too high compression, or to highly heated projections within the compression space. The latter cause of this annoying trouble has frequently been a puzzle for some of the best gas-engine men to find. A thin projection of metal within the cylinder may be so situated that it becomes heated to a comparatively high temperature and acts in the same manner as an ignition-tube. Again, there may be a projection within the cylinder upon which carbon will deposit in the shape of a cone. This cone of carbon will become incandescent, or nearly so, and cause premature ignition, even as early as on the suction stroke. Projections upon the piston head such as the heads of for lower-bolts, nuts, etc., quite frequentl make trouble in this way. In two-cycl engines of the Day type, explosions wil sometimes occur in the crank-chambe because of an insufficient fuel supply.

Flame Blown Out.—If in engines of th Otto slide valve type, the flame is frequentl blown out and there is no draft to whic this trouble may be traced, it is a sign o a leaky slide valve, and that either th springs need tightening or the valve itsel needs facing. Should the flame on a tube-igniter be blown out, there is a leak eithe in the tube itself or in some surroundin part of the engine.

Spark Gradually Weakens.—A spark tha is of the proper strength when the engin is started, may gradually get weaker afte the engine is running, until it is finally to weak to ignite the charge. If the spark i furnished by a battery, this is a sign tha the cells need recharging or that the cell i not adapted to the work. Battery cells tha are made for open circuit only, as those o the sal-ammoniac type, are unsuited fo gas-engine ignition, because they polariz rapidly and there is not sufficient time fo them to recover between the sparks. Occasionally a magneto will grow weaker after few hours use, showing that the magn are not strong enough to stand the work required of them. The remedy is to get a better magneto.

Engine Pounds.—Look the engine over carefully to determine if there are any loose bearings. Lost motion is the prevalent cause of noise in any machine or mechanism. If the bearings are "snug," note if the igniter has too much lead, or if premature explosions occur from any other cause.

Engine does not Develop Full Power.—Note what is said in the paragraph on *Weak Explosion;* then, if the trouble be not found, see if the exhaust passages are obstructed in any way. See also *Engine Slows Down.*

Smoke.—A black smoke may sometimes be observed issuing from the open end of the piston. In this case the piston is leaking. The remedy will suggest itself, upon taking out the piston and examining its condition. If the cylinder is badly out of round it should be rebored. The packing rings may need renewing. See if they are too small to expand to a size slightly greater than the bore of the cylinder. Smoke from the open end of the cylinder may also come from overheating. Smoke issuing from the exhaust-pipe, is due to an excess of fuel in the mixture.

Leaks.—To stop a leak at any point about

tion. The best remedy
stem is to ream out the bearing and
a bushing or a larger stem, being ca
see that the stem is in line when th
complete, and that the bearing cent
the valve seat.

CHAPTER VII.

GASOLINE ENGINES.

In general details and appearance, there is little to distinguish between the gas engine and the gasoline engine. The only point of difference being, that a gasoline engine has a special attachment for the purpose of supplying fuel to the engine cylinder, either in the form of vapor or a finely divided spray. In engines where the compression within the cylinder is carried to the practical limit, it is found that the limit is a somewhat lower pressure for gasoline than for gas of the average quality employed for a gas-engine fuel. A gasoline engine will develop more power than can be obtained from the same engine using a good quality of natural gas. A mixture of gasoline and air will become entirely ignited in much less time than a mixture composed of air and gas. This peculiarity of gasoline produces a much more powerful blow at the beginning of the power stroke, and usually causes the indicator to show a much higher pressure at this point than is probably present in the engine cylinder, owing to the inertia of the pencil mechanism being too great to allow it to stop when the maximum pressure is reached. A study of gasoline-engine indicator diagrams will show this effect, as illustrated in Chapter XIV.

Devices for supplying gasoline to the engine, may be divided into three classes; carbureters, vaporizers and jets.

A *carbureter* is a device for transforming liquid fuel into a vapor by passing air either over or through a body of the liquid, and carrying off a portion of the liquid in the form of vapor with the air. Carbureters usually operate at ordinary temperatures, but for fuels that have a low specific gravity the air or the fuel and sometimes both, are heated. This mixture of gas and air is usually too rich in fuel to be explosive, and a further addition of air in the engine cylinder is required before it is suited to the work.

A *vaporizer* is an appliance for transforming into vapor, just the quantity of gasoline that is required for one impulse of the engine and no more, and it differs from the carbureter in not having a supply of vapor constantly on hand. Either the proper

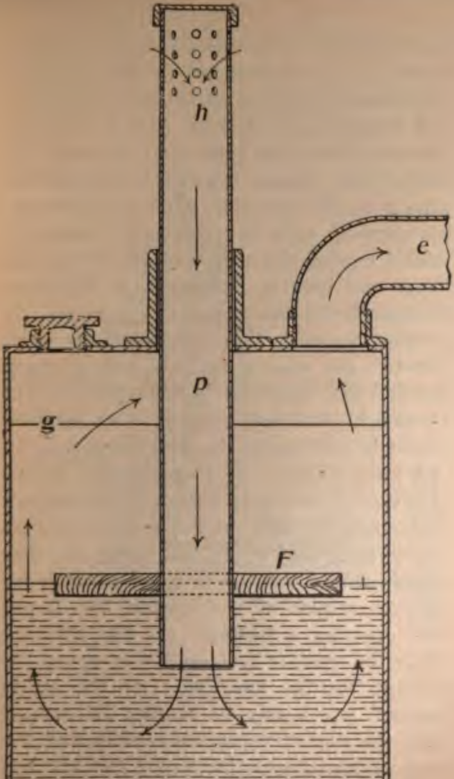


Fig. 5.

quantity of fuel is caused to flow directly into the path of the entering air, or the air is passed over a pipe connecting with a small gasoline reservoir and a current of the fuel is induced into the path of the entering air.

Jets are what the name implies, a jet of liquid usually controlled by a small pump. The pump throws a jet of the liquid into the air pipe so that it strikes the side of the pipe and breaks into a spray, or, as in certain classes of kerosene engines, into a compartment of the compression space and against the side. Jets are sometimes classed as vaporizers, but placing them in a class by themselves makes them much more convenient to refer to.

Carbureters may be divided into two classes, *surface carbureters* and *filtering carbureters.*

In Fig. 4 is shown an example of a surface carbureter. The carbureter is constructed in the form of a spiral in order that the air passage through it may be a long one. The bottom of the carbureter is covered with gasoline to the height xy, and the wicking ww absorbs the liquid so that a large surface of fuel is exposed to the air as it passes through. According to Mr. Gardner Hiscox, the height of the gasoline should be not over 3 inches and the total height of

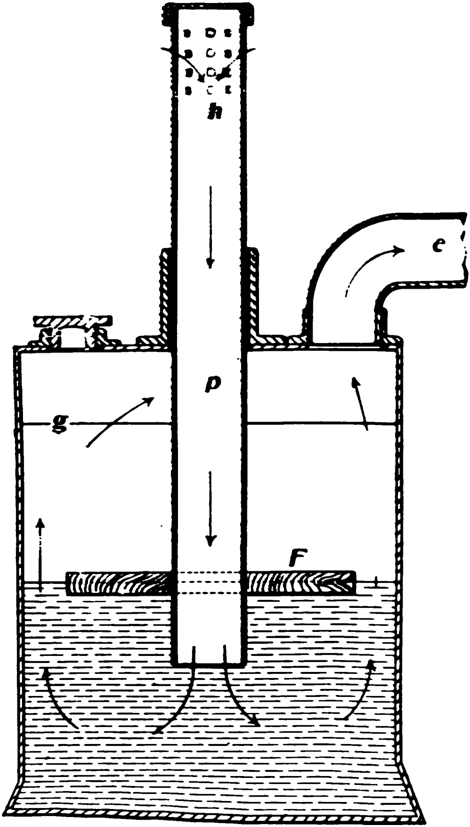


Fig. 5.

CHAPTER VIII.

HANDLING A GASOLINE ENGINE.

For starting, stopping and the care of a gasoline engine, the same general rules apply as for a gas engine. In starting a gasoline engine, especially when the engine is cold after standing idle for some time, it is a good plan to put a quantity of gasoline in the cylinder and allow it to remain there for about a minute before starting the engine. With engines employing a pump to raise the gasoline from a tank or reservoir, it is necessary to operate the pump by hand for a few strokes in order to get a supply of fuel in the reservoir. The fuel supply to the engine is usually regulated by means of a needle valve, which should be carefully cleaned at regular intervals. In engines using a jet feed, the supply is regulated by adjusting the stroke of the pump, or by regulating the opening in a by-pass, so that a portion of the fuel is pumped through the by-pass and returns to the source of supply.

With engines using a carbureter, it may be found necessary, in extremely cold weather, to arrange some means of supplying heat to it, because the transformation of the fuel into vapor produces a refrigerating effect which will chill the liquid to such an extent that it will not vaporize. Two methods of heating the carbureter may be employed; one is to pass the exhaust-pipe through the liquid and the other is to warm the fuel by means of the outgoing jacket water. The latter method is undoubtedly the most satisfactory as it avoids excessive heating of the gasoline. In kerosene-oil engines which operate on the carbureter or vaporizer principle, it is found absolutely necessary to heat the fuel before it enters the cylinder. In some forms of jet supply, it is not necessary to heat the kerosene before it enters the cylinder, as it is either injected against a highly heated surface or into a body of air that has been brought to a high temperature by compression, as in the Diesel motor.

The time of ignition in the gasoline engine may be made a little later in the cycle, and thus avoid the hard blow produced at the time of the explosion, without a noticeable loss in power. In starting a gasoline engine, it is often necessary to *first keep the* regulating valve entirely closed and to open it a little at a time immediately after the engine receives its first impulse. In general, a carbureter will produce a more perfect mixture than the jet or the vaporizer, thus insuring complete combustion and a consequent absence of smoke and odor at the exhaust. But this advantage is, in many engines, offset by the many disadvantages of this method.

The carbureter usually takes' but the lighter portions of the fuel, leaving in the bottom of the tank a residue which is of little or no value, as it is of too low a specific gravity to be used as a fuel in the engine. On the other hand, the vaporizer or the jet insures the using of all the fuel irrespective of specific gravity, if the average of the mixture is equal to that required by the engine. In the occasion of the engine being located near a refinery, it is probable that the residue resulting from the use of a carbureter may be sold at a price that will overcome that objection to this method. In general, however, the vaporizer or jet will be found the least troublesome, although some very successful engines are employing carbureters.

At no time should there be any foreign matter permitted to enter the gasoline-supply pipes or the valves. Gasoline tanks should be filled through a fine-wire strainer or a piece of closely woven muslin. The opening of the supply-pipe into the tank should be covered with a strainer and every precaution taken to prevent trouble from this source. Never put the gasoline in anything but a perfectly clean receptacle. Old paint or varnish cans or barrels are especially to be avoided. At regular intervals, drain the gasoline tank and clean it thoroughly. Where the tank is under ground or in a similar position, in which it would be impractical to clean it, extra precaution should be taken to avoid the entrance of foreign matter. Much of the trouble experienced with gasoline engines occurs from neglect of this precaution.

Large quantities of gasoline should be stored at some distance from buildings, and the tank should be protected from the direct rays of the sun by means of a shed or a covering of earth. If the tank is made of sheet iron and fitted with a safety valve that will allow the escape of the vapor, so that the pressure induced by overheating may be relieved before endangering the tank, much of the loss by evaporation will be avoided. Many gasoline-tank explosions are due to excessive pressure caused by the overheating of a tightly closed receptacle and not from the application of a flame *Usually, the* evaporation of this ligh hydrocarbon is so rapid that all air is driven from the vacant space above the liquid and a flame applied to an outlet of the tank would not cause an explosion, as the mixture would be too rich.

If, by any mishap, a tank of gasoline takes fire at a small outlet, run to the tank and not away from it, and either blow or pat the flame out. Never put water on burning gasoline or oil, for the oil will float on top of the water and the flames spread so much the more rapidly. Throw fine earth, sand or flour on top of the burning liquid. Flour is best, because it will float and less will be needed. The best fire-extinguisher for a fire of this sort in a room that may be closed is ammonia. Several gallons of ammonia, thrown in the room with such force as to break the bottles which contain it, will soon smother the strongest fire if the room be kept closed. Very often, simply striking the opening from which the flame is issuing, with the palm of the hand will put out the fire.

CHAPTER IX.

IGNITERS.

A good igniter is one of the most important parts of a gas engine. It is required of an igniting device: that it shall fire every charge without fail, that it shall ignite the charge at the proper point of the cycle, and that it shall require a minimum amount of attention both for cleaning and renewal of parts. There are four distinct ways in which the charge may be ignited

I. Ignition by means of a naked flame.
II. Contact with a surface which is at high temperature.
III. The flame of a small electric arc.
IV. Raising the temperature of the cylinder contents by high compression.

Flame ignition has its best known example in the Otto slide-valve engine. The general principle involved is, however, much better illustrated in Barnett's igniting cock, Fig. 7. In this method of ignition two gas jets *are necessary*. One, the flame *f* which is

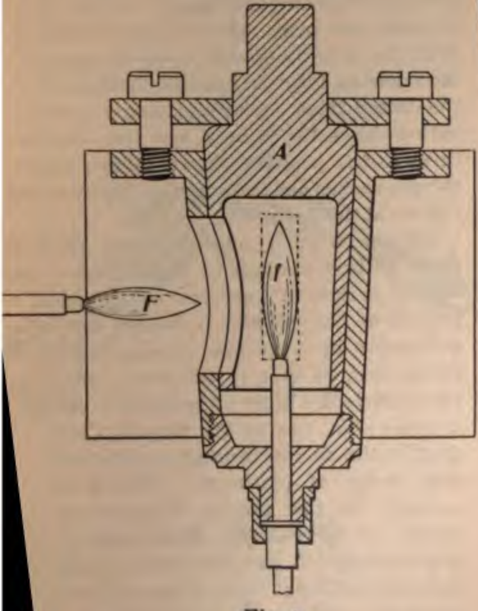


Fig. 7.

employed for ignition of the charge, and the other *F* for relighting *f* when blown out by the explosion within the cylinder. The plug valve *A* is shown in the proper position for relighting the flame *f*. At the right moment, the valve makes a quarter turn so that the opening in the plug is opposite the opening into the cylinder indicated by the dotted lines. The resulting explosion extinguishes the flame *f*, and the valve returns to the position shown in the figure, the gas rushes out through the opening and ignites at the flame *F*.

Contact with a highly heated surface finds its best example in the hot tube. In several makes of kerosene oil engines, a portion of the cylinder is divided from the remainder by a narrow passage and left unjacketed, so that it reaches a temperature sufficiently high to ignite the charge. Another plan, due to Mr. Dugald Clerk, is to drive the charge through a grate built of thin strips of platinum. The grate is brought to the proper temperature by means of an exterior flame before the engine is started and, thereafter, the grate receives sufficient heat from the burning gases to ignite the following charge. This method is useless in a hit-and-miss type of engine, because a few idle strokes will allow the grate to cool to a temperature below

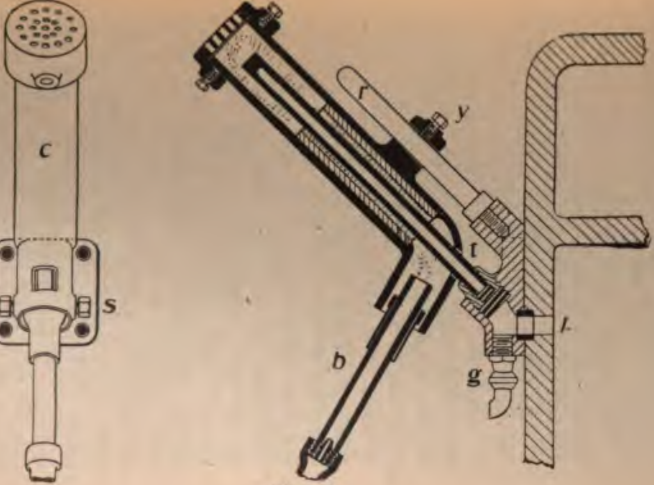


Fig. 8.

that necessary for ignition. Thin rods and even small bolts have been employed in much the same manner as the platinum grate.

The earlier tube igniters used what is known as a timing valve. This valve opened communication with the tube, so as to time the ignition at the proper point of the cycle. It is now the common practice to time the firing point automatically by the compression of the charge. When the engine exhausts, the pressure within the tube falls to that of the atmosphere, and it is filled with the products of combustion. When the engine compresses a fresh charge, a portion of the mixture is driven into the ignition tube, forcing the products of combustion ahead of it, and when the pressure within the cylinder has reached the right amount, the fresh mixture is brought into contact with the heated portion of the tube and ignites. A tube of this kind with adjustable burner, is shown in Fig. 8. The tube is shown at *t*, and it should be of either nickel alloy or porcelain. The gas is driven by the compression of the piston into the tube through the port *p*. The tube is heated by the Bunsen burner *b*, the flame entirely surrounds the tube and is confined by means of the chimney *c*. The chimney is lined with a tube of asbestos, the asbestos being an important addition, as it prevents the heat from being carried off through the walls of the chimney. Minute adjustments of the flame may be obtained by swinging the burner *b* about the set-screws *s*, while for larger movements the chimney and the burner may be changed by loosening the screw *y* and sliding the chimney along the rod *r*. A pet-cock is provided at *g* for blowing the soot from the tube. In general the flame and the chimney are not made adjustable, their proper position having been determined by experiments at the factory, and the chimney and burner being fastened in the position found most favorable for the operation of the engine.

Ignition by contact with a hot surface in the combustion chamber, is illustrated in Fig. 9. This set of diagrams shows the series of operations which take place in the Hornsby-Akroyd oil engine. Before starting the engine the chamber *c* is brought to a temperature very nearly that of a red heat and this temperature is afterwards maintained by the combustion within the cylinder. During the suction stroke of the engine, a jet of oil is forced into *c* by means of a pump and, striking the hot surface of the chamber, it is transformed into vapor. The cylinder of the engine when the piston is at the end of the suction stroke contains

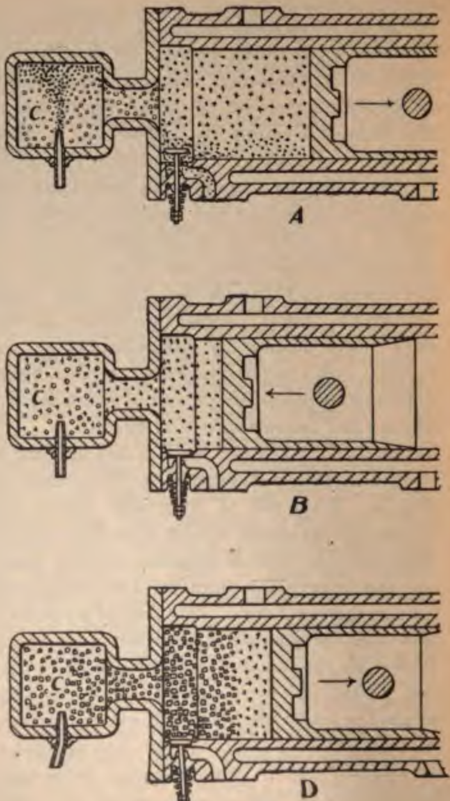


Fig. 9.

pure air, while the chamber *c* is filled with oil vapor and products of combustion left from the last cycle. In the figure, the oil is represented by small circles and the air by crosses, while the products of combustion are shown by small squares. In diagram (B) the piston has compressed the air, driving it into *c* and, as soon as the contents of the chamber is of an explosive nature, it takes fire from the heated surface of *c*. In diagram (D) the piston has started on a forward stroke, expanding the products of combustion.

The electric igniter is slowly but surely taking the place of all others, because only by its use can the ignition of the charge be timed to a certainty. The form of electric igniter which is in most general use, operates upon the following principle: An electric circuit from a battery or other source of electrical energy is closed, by means of contact points within the compression space, through an inductive resistance in the form of a spark coil. Upon breaking the circuit, the inertia produced by the induction raises the pressure of the circuit and causes a hot spark to arc across the terminals. This method is known as the make-and-break, and it may be produced either by forcing the two contacts together and then throwing them suddenly

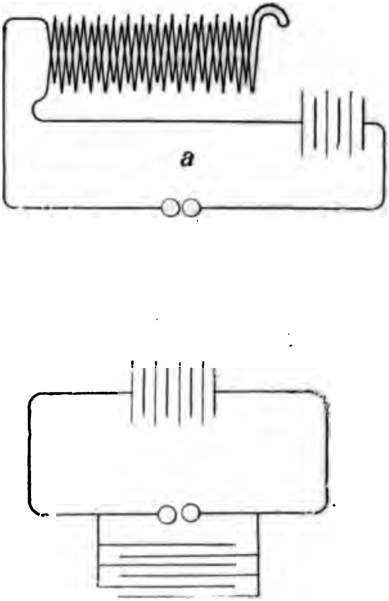
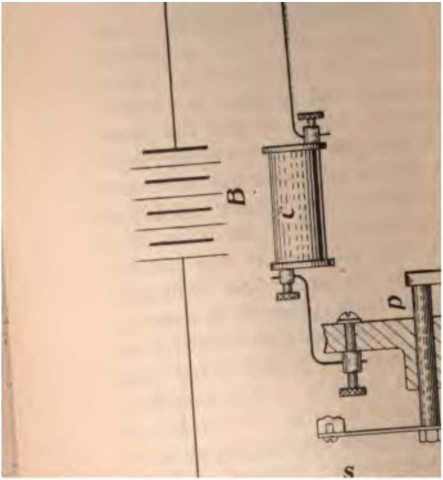


Fig. 10.

apart by means of a spring, or by wiping one contact on the other in a manner that is known as the wipe break. This latter method produces a very hot spark, but provision must be made for adjustment as the points wear out quite rapidly.

In either method, good judgment must be used in adjusting the current supply to the requirements of the ignition device. Too little pressure will produce but unsatisfactory ignition, while, should the pressure be such that the current will be large, the contact points will wear out in a very short time. Where the battery employed is of low resistance, as in the case of a storage cell, the pressure at the battery terminals should be much less than with cells having a high internal resistance. Two methods are available for reducing the rapid destruction of the points by too large a flow of current, when it is inadvisable to reduce the number of cells. The destructive action of the spark may be annulled by placing a condenser in parallel with the break as shown in Fig. 10, or by putting a *noninductive* resistance in series with the circuit as shown in Fig. 10a. To make a noninductive resistance, wind the wire in a coil, about a core that contains no iron, and begin to wind at the center of the length of wire *so that*, when finished, one half the


B
C
p
s

current will flow in one direction around the coil, and the remaining half in the opposite direction. If the resistance coil is not made noninductive, it will be quite sure to destroy the sparking power of the spark-coil.

An example of the first style of make-and-break is shown in Fig. 11. This figure is diagrammatic only and is not intended to represent any special make of igniter. From a source of energy *B*, one wire is grounded on the frame of the engine. The other side of the circuit is attached to the insulated electrode *e*, and it contains the spark coil *c* connected in series. The cam *C*, rotated in the direction of the arrow, depresses the spring *s* which carries with it the electrode *p* until the point of *p* meets *e*. Further rotation of the cam merely deflects the spring, increasing the pressure between the contacts. Still further rotation of the cam allows the end of the spring to slip off the lip and it flies back, carrying with it the plunger *p*, making a quick break between the contacts and producing the spark. In practice, much trouble is experienced with flat springs as they are more liable to break than are helical springs. Note that by helical spring is meant a spring that is wound in the form of a screw thread as shown at *A*, *Fig.* 12. This form of spring is

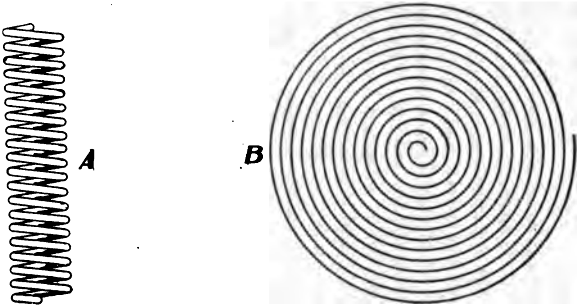


Fig. 12.

often but erroneously called a spiral spring. The true spiral spring is shown at *B* in the figure, and is of the kind so much used in clocks; it is not so reliable as the helical spring.

An example of a wipe break is shown in Fig. 13. The electrodes *X* and *Y* are so placed that they lie directly in the path of the incoming gases in order to keep the electrodes cool. The electrode *Y* is rotated by the crank *E*, and rubs against the spring electrode. The rubbing has the advantage that it keeps the contact surfaces always clean. The electrode *Y* is made adjustable, so that it can be pushed further into the valve box as it wears off. A very hot spark is produced by this form of break and, although it requires more frequent adjustment and renewal of parts than some other forms, there is no platinum used, so that it is quite a favorite with some designers. It should be borne in mind by the designer that, in both these forms of make-and-break igniters, contact must be made with pressure and the break must be a quick one.

Another method of electric ignition that is finding great favor, especially among automobile builders, is the jump-spark. This spark is so called because it will arc or "*jump*" *across* an air gap without pre-

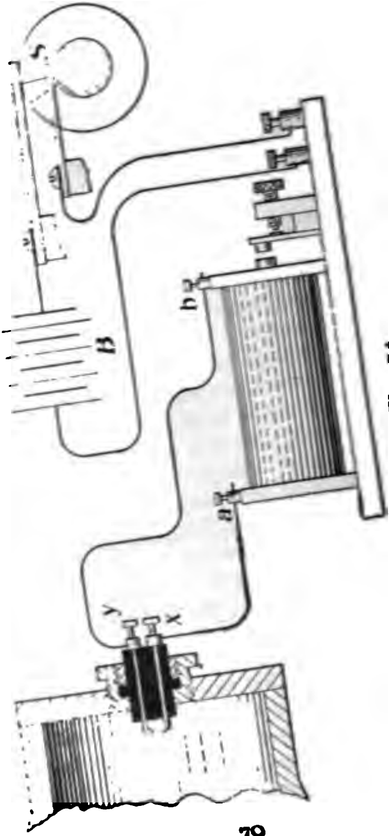
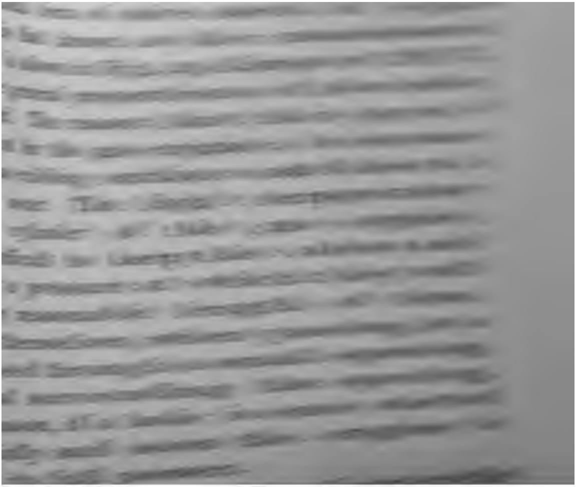


Fig. 14.

switch. The second spark is that which is depended upon for igniting the charge, as the make spark is not strong enough.

The arrangement of the circuits for producing the jump-spark is shown in Fig. 14. The primary circuit passes through the switch *S* on the cam shaft to the primary of the coil, through the primary and thence back to the source of energy *B*. The secondary circuit is from the terminal *a* of the secondary to the terminal *x* of the ignition plug and from the terminal *y* of the ignition plug to the terminal *b* of the secondary. For the secondary circuit, the secondary winding of the coil becomes the source of energy. If there is a vibrator in the circuit, the time of ignition will be immediately after the switch closes the circuit. If there is no vibrator in the circuit, the time of ignition will be just as the switch breaks the circuit. To get a sure ignition, it has been found that the points should be set a distance apart equal to about one quarter the maximum sparking distance of the coil. A ½-inch coil with a ⅛-inch gap has been found to give very good results. There are coils on the market designed expressly for gas engines using the jump-spark, and the author strongly advises the reader to buy a coil rather than attempt to make it himself, for the reason that considerable skill and

description of this method the
referred to Chapter I.



is employed which is on a shaft making one revolution to four revolutions of crankshaft. The valve ports are so arranged that they are open to the cylinder twice in each revolution of the valve-shaft.

In many small engines, only the exhaust valve is opened by mechanical means, the inlet valve being operated by the suction of the engine. This of necessity causes some wire-drawing during the suction stroke, as there must be a partial vacuum in the cylinder before the valve will operate. On large engines, wherein a small proportionate loss of power becomes of importance, it is not advisable to employ the suction valve, and the inlet valve should be operated in the same manner as the exhaust valve. In fact, some gas-engine

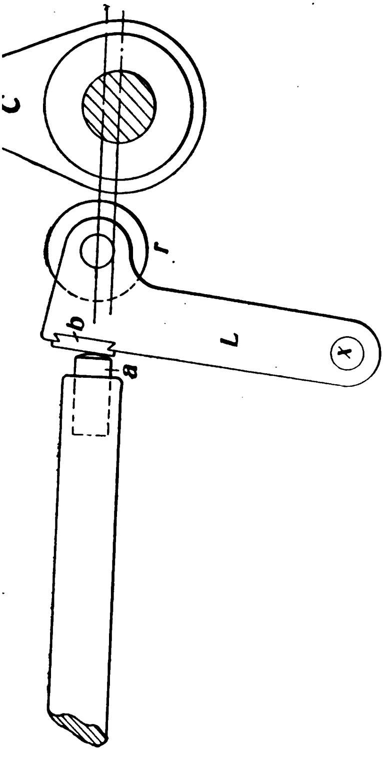


Fig. 15.

are, however, some very good engines and a few large ones which have been operating for several years with suction valves. The only advantage the author can see in such a valve is simplicity and less first cost, because of the absence of the valve mechanism.

In two-cycle engines of the Day type, the only valve required for either the exhaust or the inlet, is a check-valve opening into the crank-chamber. For a description of this engine and the valve arrangement the reader is referred to Chapter I. The check-valve will cause a slight wire-drawing in the crank-chamber, but, with a properly proportioned engine, there is no vacuum formed in the cylinder at any portion of the stroke.

It is customary, in the design of a valve mechanism, to transmit motion from the cam to the valve-stem by means of a lever in order to avoid cramping of the valve-stem in its bearings. If, however, the valve-stem be given a bearing near the cam with a long surface and the stem be enlarged if necessary in order to stiffen it, the intermediate lever may be omitted. A valve-operating device with a lever is shown in Fig. 15. The lever *L* is pivoted at *x* and carries the roller *r* which bears against the *cam C*. The center of gravity of the le-

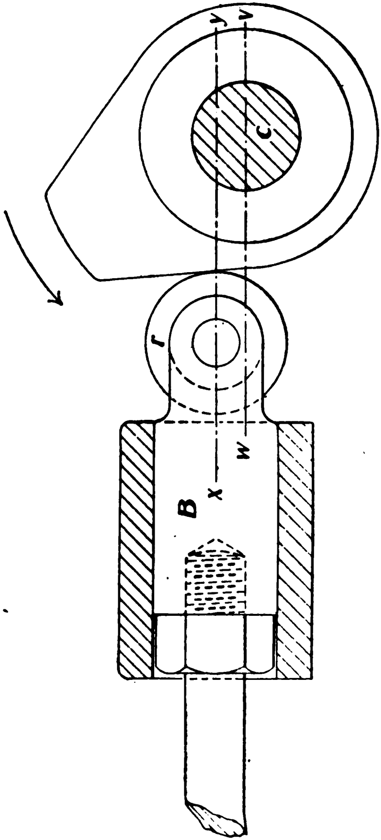


Fig. 16.

L is placed far enough to the right of *x* that the lever will incline toward the cam after it has been thrust by the valve-stem *s* as far as allowed by the head of the valve. The hardened steel contact piece *a* is put in the end of the valve-stem and a hardened plate *b* is dovetailed into the lever.

In Fig. 16 is shown a valve-stem without the lever. The stem is screwed or keyed into the sliding block *B*, which is forked on the end so as to provide a bearing for the shaft of roller *r*. If the camshaft rotates constantly in one direction, the center line of the camshaft should be set out of line with the roller shaft as shown in the figure. The direction in which the shaft *c* should be out of line is determined by the direction of motion of the camshaft. The line *w v* should be on the side of *x y* opposite the projection on the cam, when the projection is approaching the roller.

In laying out a cam, the simple method shown in Fig. 17 will give an outline that is suitable for nearly every condition. The cam is made up of two parts, a portion *C* concentric to the shaft, and the eccentric portion *P*. The part *C* is usually turned a little smaller than the circle shown by the dotted line, which is a circle that would be described by the roller when the valve is down upon its seat and all lost motion

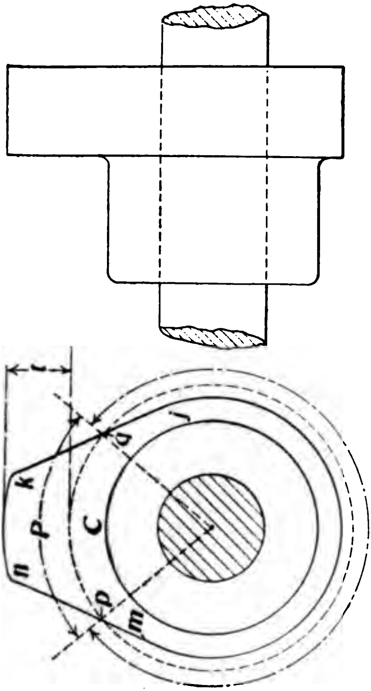


Fig. 17.

is taken up between the stem and any mechanism which carries the roller. In laying out the eccentric portion, it should be remembered that the valve does not begin to open until the point *p*, where the line *m n* meets the dotted circle, is in contact with the roller. That portion of the revolution of the cam during which the valve is to remain open, should, therefore, be laid off on the dotted circle and not, as the author has seen it done, on the concentric outline of the cam. The arc *P* is usually so laid out for the exhaust-valve, that it will open the valve when the piston has completed about .9 of the working stroke and close it just as the exhaust stroke is completed. Hence, for the exhaust cam the arc *P* should be 1.1 times a quarter circle or 99°, say 100° in round numbers. After laying off 100° on the dotted circle, draw tangents through *p* and *q* to the circular outline of the cam, meeting it in the points *m* and *j*. Lay off the radial distance *t* equal to the movement of the roller when it is operating the valve, and describe the arc *n k* slightly rounding the corners at *n* and *k*.

For the inlet cam the arc *P* is usually slightly under 90° in order that it may not open the inlet valve before the piston starts on the suction stroke nor hold it open after the compression stroke has begun. The

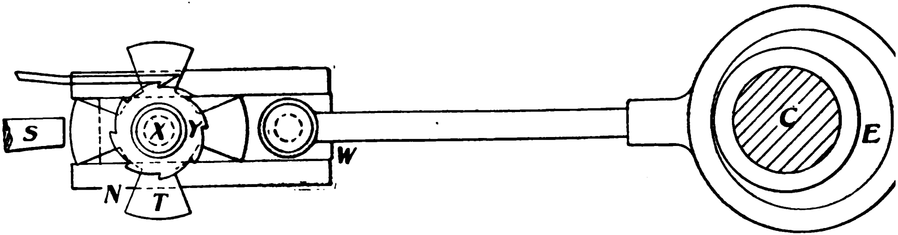


Fig. 18.

designer should exercise his judgment to some extent in this matter, but he will find 85° to be very close to the proper length of the arc. For the gas valve, a great many designers use the same cam as for the air valve. In any case, the time that the gas valve should remain open is practically the same as for the air valve, some engines being governed within limits by varying the time during which the gas valve remains open.

An example of a valve operating mechanism in which there is no camshaft, is shown in Fig. 18. In the figure, *s* is the exhaust-valve stem. The slide *W*, reciprocated by the eccentric *E* on the crankshaft *C*, carries a toothed wheel *X*. To the wheel *X* is pinned the ratchet wheel *Y* which has just twice the number of teeth that are on *X*. At each stroke of the slide *W*, made in the direction of the crankshaft, the pawl *p* rotates the wheels *X* and *Y* so that a tooth *T* and a notch *N* are presented alternately to the stem *s*. Hence the valve is opened but once during two revolutions of the engine. It is claimed by those manufacturers who employ this and similar devices, that the valve opens much more quickly than with the cam, and a better performance of the engine is obtained in consequence. There are several varieties of valve-motion dependent upon an eccentric and a device similar to that shown in Fig. 18, nearly all of which are the same in principle as the device shown. A number of devices for avoiding the two-to-one reduction have been patented, a few of which are in use, but the majority of the four-cycle engines are using the camshaft.

For transmitting motion from the crankshaft to the camshaft, there are three kinds of gears employed—the ordinary spur gear, the bevel gear and the skew gear. The spur gear is that which has its teeth on the periphery of a disk, and is used for transmitting motion between shafts which are parallel. It is the most familiar of the three. The bevel gear has its teeth on the surface of a cone and is used for transmitting motion between shafts at an angle but which lie in the same plane. The skew gear is one in which the teeth are in the form of a screw thread or helix—a screw with a multiple thread—and is used for transmitting motion between shafts which are at an angle and which do not lie in the same plane.

Examples of these three varieties of [illegible] are shown in Fig. 19. At (A) is shown a pair of spur gears for transmitting [illegible] from the crankshaft s to the [illegible] and at the same time reducing the [illegible]

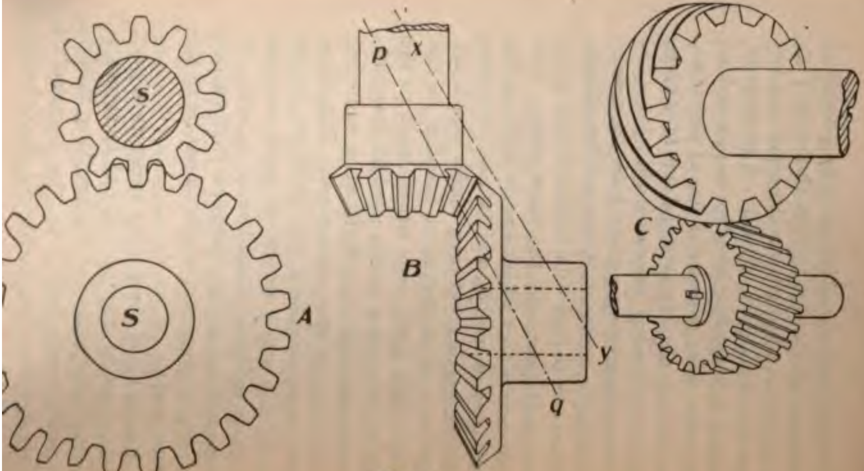


Fig. 19.

S to half that of s. The size of pinion p is made one-half that of the gear G, in order to make the proper reduction in speed. The two shafts are parallel. At (B) is shown a pair of bevel gears for changing the direction of motion and at the same time making the two-to-one reduction. The shafts lie in the same plane, that is to say, straight lines, as $x\ y$, $p\ q$, etc., may be so drawn as to pass through the centers of the two shafts. At (*C*) is shown a pair of skew gears in which the speed reduction may be made without making the slower rotating gear the larger. In fact, the gear on the camshaft may be, and usually is made smaller than that on the crankshaft. Another advantage is that the two shafts do not lie in the same plane. Motion may be transmitted with these gears without the noise that is more or less evident when the spur or the bevel gear is used.

Engines, which are fitted with spur gears, usually have the camshaft near the crankshaft, and the valve-stems project from the valve boxes in a direction parallel to the axis of the cylinder. This arrangement necessitates either long stems on the valves, or a long rod from the camshaft to the end of the valve-stem In some engines, this feature is avoided by the use of a train of spur *gears which* brings the camshaft close to the valve boxes. In a carefully designed engine, there is no objection to the use of spur gears, and their use often saves expense in construction.

Very few engines employ the bevel gear, as it is a somewhat difficult matter to so arrange them on the engine that they will not take up an undesirable amount of room. The skew gear obviates this latter difficulty to a great extent, and it allows the camshaft to be placed in a position from which it is more convenient to operate the valves. Either gear may be used to operate a camshaft the axis of which is parallel to the axis of the cylinder, and by the use of the second pair of gears the shaft may be given a second quarter turn and run across the back of the cylinder head to operate valves that are placed in the head. Where there are two or more cylinders, it will add to the economy of construction, if a single camshaft is employed to carry all the cams. This shaft should then be placed so that the axis is at a right angle to the axes of the cylinders.

CHAPTER XI.

GOVERNORS.

All gas engines are governed by either cutting out or cutting in impulses to the piston, or by reducing or increasing the force applied the piston according to the requirements. Regulating the speed of the engine by varying the number of impulses given to the piston is called the hit-and-miss system. There is no term at present in use that will cover all cases of governing by varying the strength of the impulse, and the author suggests the term *variable impulse.* There are three methods of governing under each of the above two systems of regulation, as described below.

HIT-AND-MISS.

I. *Holding the gas valve closed* during one or more revolutions of the engine when running below full power. During the idle strokes the engine is compressing and expanding a charge of pure air.

II. *Stopping the action of the exhaust valve*, holding it either open or closed during the idle strokes. When the valve is held open there is not sufficient suction to open the inlet valve, this method usually being employed on engines having a suction valve. In the latter case the engine retains the products of combustion, the pressure within the cylinder being at all times too great to allow a fresh charge to enter.

III. *Cutting off the current from the igniter*, employed, of course, where an electric igniter is used. The governor is, in this case, attached to a switch, which is opened whenever the speed passes the limit. In this case the charge is alternately compressed and expanded until the switch is closed and ignition takes place.

A fourth method might be added to this class, *i. e.*, the stopping of the camshaft. With this method all operations are halted, usually with the exhaust valve held open.

VARIABLE IMPULSE.

IV. *Partial stoppage of the gas supply.* The mixture of fuel and air retains its explosive properties within certain limits of the proportion between the two. If the mixture is poor in gas the impulse is correspondingly weakened, and up to the point ereat the charge explodes with the reatest force, the strength of the impulse may be increased. There is also a lower limit, beyond which the mixture becomes so poor in gas that it will not explode. After passing this limit it is necessary to resort to the hit-and-miss method I, and cut out the gas supply entirely.

V. *Throttling the charge.* This method reduces the strength of the impulse by reducing the quantity of the charge that enters the cylinder, the proportions of fuel and air remaining the same at all times.

VI. *Varying the point of ignition.* The greatest mean effective pressure in a gas engine is obtained when the spark has sufficient lead to bring the maximum pressure at the beginning of the stroke. If the lead is reduced, or made negative (ignition delayed until the crank has passed the center), the strength of the impulse may be varied.

Hit-and-miss regulation is that which gives the greatest economy of fuel consumption. There is little choice between the three methods given above, as all have advantages and disadvantages that make them about equally good.

Of the variable impulse methods, No. IV is that which gives the best fuel economy, as the *compression* is the same at every cycle. The objection to this plan is, the return to the hit-and-miss system as soon as the speed falls to the limit below which the charge would be too poor in gas to explode. Method V has been used quite successfully for engines driving dynamos, But it does not give the economy that is derived from an engine employing one of the first four methods. The method described last, is perhaps the most wasteful of fuel after the lead is late enough to form a perceptible reduction in the area of the indicator diagram. The result obtained is shown very plainly in the chapter on indicator diagrams. This method of speed regulation is of little service where economy is desired, but where economy is a secondary consideration, as in an automobile engine, it is a very simple method of governing.

Governors of three different kinds are employed on gas engines. Two of these are of the centrifugal type, similar to the governors in use on the steam engine, the other is a governor made possible by the hit-and-miss system of regulation and is known as the pendulum or inertia governor. The centrifugal governors are the conical pendulum and the shaft governor. Either of these two may be employed for any method of regulation while the pendulum

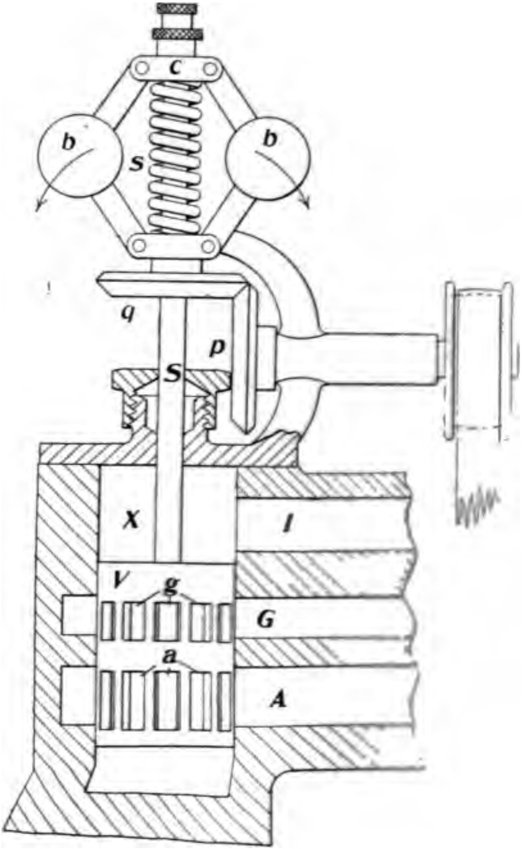


Fig. 20.

governor may be used only for the hit-and-miss.

An example of the conical pendulum governor is shown in Fig. 20 as arranged to operate a throttling device. The two balls *bb* are rotated from the camshaft, or other convenient revolving portion of the engine, through the bevel gears *pq*. As the speed of rotation increases, the balls move in the direction of the arrows, compressing the spring *s* and depressing the stem *S* of the valve *V* by means of the collar *c*. As the valve moves downward, the holes *a* and *g* are gradually covered by the wall of the passage, contracting the opening from the gas passage *G* and the air passage *A* into the chamber *X*, and preventing the mixture from flowing in as great a quantity as before, into the inlet port *I*.

An example of the shaft governor is shown in Fig. 21, in connection with a device for opening the gas valve only when the engine is below speed. The balls *bb* are on the end of the short arms *ll* pivoted at *pp*. As the crankshaft *C* revolves, the balls tend to swing in the direction indicated by the arrows, but they are withheld by the springs *s*. As the balls swing about the pivots *pp*, the opposite ends of the arms *ll* move toward *X* and carry with them the collar *c*. A long lever *m* with a fulcrum pivot at P is

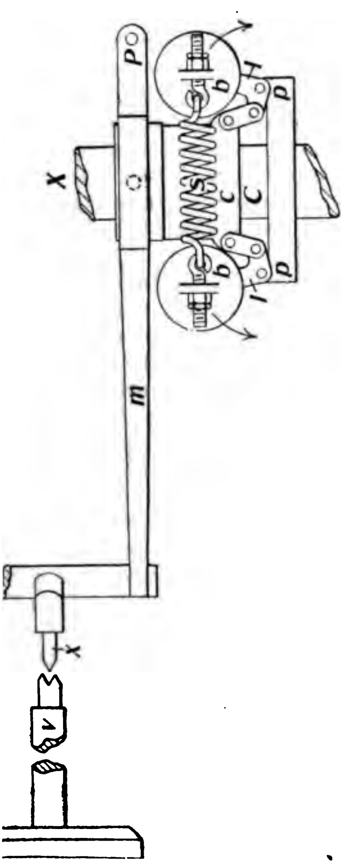


Fig. 21.

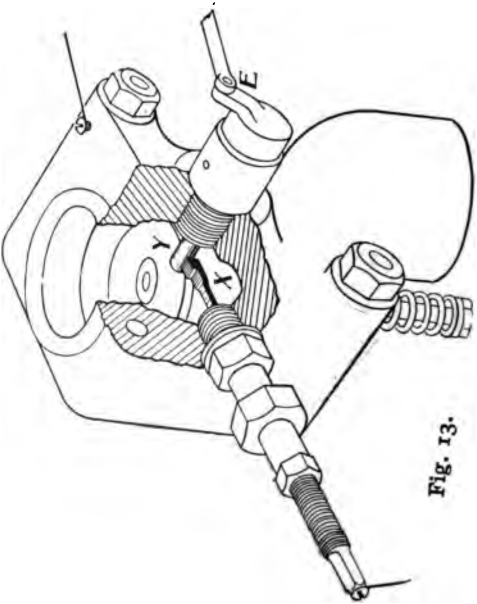


Fig. 13.

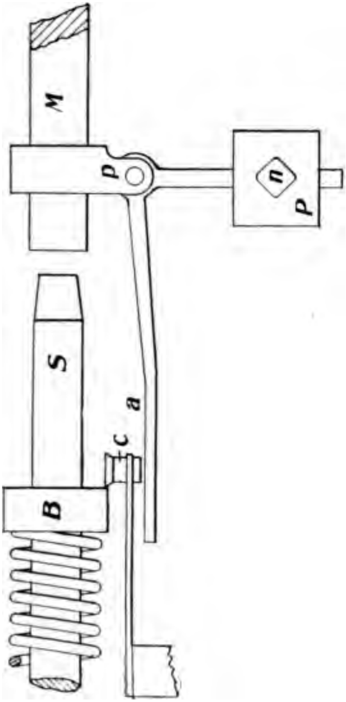


Fig. 22.

to control the gas valve.
of the bell-crank is pointed a
to give the necessary thrust
valve. In case the speed rise
tain amount the arm *a* swin
misses the gas valve stem.

CHAPTER XII.

STARTERS.

Engines larger than eight or ten horsepower are difficult to start by turning the flywheel by hand, and, in order to lessen the amount of labor involved, larger engines than the above are usually supplied with some device by means of which an impulse may be given to the piston when the engine is at rest, that will be strong enough to propel the engine for three or more revolutions and until it can take up its cycle and receive an impulse in the regular way. The methods in use for this purpose are so numerous that but a few examples may be illustrated in a volume of this nature.

The first method to suggest itself to an engineer would obviously be to apply pressure to the engine for one or more strokes from an exterior source, as from a steam boiler or from a tank of compressed air. In fact, the compressed air starter is a most pounds or over, when the

the pump being operated by means of a belt from the line shaft or other convenient point. When it is desired to start the engine the flywheel is turned until the camshaft is in the proper position for the exhaust stroke, and until the crank-pin is just above the back-center. The air-valve is then opened wide, and closed again as soon as the piston nears the end of its stroke. If the impulse is not strong enough to give the engine a sufficient start it should be repeated at the next revolution but one. Two impulses with the air are usually more than enough for the purpose. After the engine is under way, the fuel should be turned on as directed in the chapter on Starting. If the engine ha

cylinder one cylinder ma

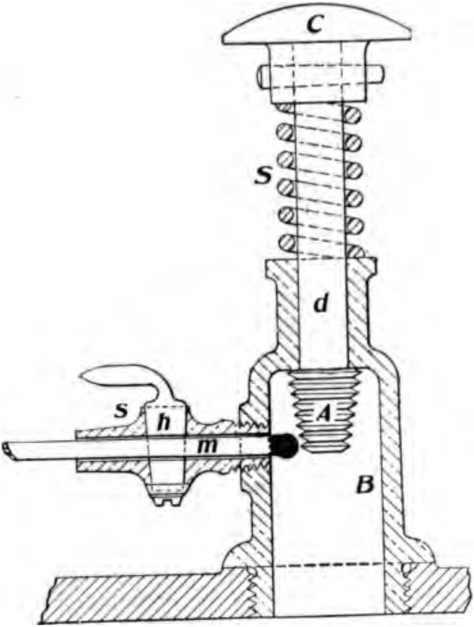


Fig. 23.

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revolution until the speed is sufficient to take up the cycle.

A class of starters that is very much in use on engines of fifty horsepower and below, is simply a means of igniting a charge that has been drawn into the engine by turning it over by hand. For instance, a certain manufacturer equipped his engines with what is called a match igniter. This igniter is shown in section in Fig. 23. The igniter consists of the casting *B* which is screwed into an opening into the valve-box. It contains a plunger consisting of a tapered and roughened head *A*, a stem *d* and a cap *C*. The plunger is held at the top of its stroke by means of the spring *S*. A small pet cock *s* is screwed into the side of *B*, through which is inserted a parlor-match *m*, the plug *h* is then turned so as to break the match-stick and hold it in place. When the engine is under way the cock *s* is opened and the burnt match expelled by the next explosion within the engine.

To start the engine, it is turned over until the igniter snaps, and a charge of air is drawn in by turning the flywheel. At the same time, a small quantity of gasoline—the amount depending upon the size of the engine—is drawn in with the air. The engine is now turned over until the piston is at nearly the end of the stroke; the engine s then reversed compressing the charge by a quick backward turn of the flywheel, the cap *C* of the igniter is struck with the hand lighting the match *m* and firing the charge. The explosion is usually of sufficient force to turn the engine until it can take up its cycle.

In some engines, the use of the match igniter is avoided by means of a device which operates the electric-igniter at the proper moment. When the jump-spark is employed, a switch which will close the primary circuit, is all that is necessary. In other engines the flywheel is turned until the camshaft is at the proper position for the beginning of the suction stroke, and the crank-pin is on the inner center. A charge of fuel and air is then pumped into the cylinder by hand. The pressure within the cylinder can be determined by the force necessary to work the pump or by setting the engine a little past the center and holding the flywheel, noting when it begins to pull. When the required pressure is obtained, the flywheel should be turned until the pin is about 10° past the center and the charge fired by means of a match igniter or equivalent device.

In large engines, where the horsepower is 100 h. p. or above, a small gas engine, which *may* be easily started by hand, is

sure tank by permitting a portio
expanding charge to pass into t
This is accomplished by opening co
cation with the tank after the pis
made a portion of its expansion str
first portion of the stroke being
impart motion to the engine.

CHAPTER XIII.

ENGINES FOR AUTOMOBILES.

The requirements of an automobile engine are lightness, compactness, minimum vibration, wide range of speed under control of the operator, and simplicity. Economy of fuel consumption is a secondary consideration. The fewer the cams, levers, etc., on an automobile engine, the less liable will it be to get out of order. It was formerly supposed by the gas-engine builders that a gas engine must run at a constant speed, and all changes of speed made by means of gears. Recent experiments have exploded this theory and successful engines are now on the market which will run at speeds varying from 100 to 1000 r. p. m. for the same engine. At least one manufacturer claims that his engine may be run at speeds ranging from 100 to 2000 r. p. m. Two methods are employed for governing the speed of the engine. One is by throttling the charge, and the other by changing the lead of the spark. Either of these methods gives an impulse at each cycle. Engines operating on the hit-and-miss principle, are useless for motor vehicles.

For single-seated carriages, an engine capable of giving at least four horsepower at between 500 and 600 r. p. m. should be employed. If the carriage is likely to be called upon for heavy hill-climbing or racing speeds the horsepower should be increased to six. For motor bicycles one horsepower is usually sufficient, while for tricycles two horsepower is that generally employed. Single-cylinder engines will not run in a horizontal position without causing an annoying amount of vibration to be transmitted to the vehicle. If it is desired to use a single-cylinder engine it should be placed with the cylinder vertical and the engine very carefully balanced. Two-cylinder engines with the cylinders placed end to end, each connecting-rod being attached to a separate crank-pin and the crank-pins placed 180° apart, will, if all parts are carefully made of the same weight, run with very little vibration. The reciprocating and rotating parts of the two engines balance each other. Two cylinders with their axes *parallel* and the open ends in the same *direction*, work very well when the cranks *are set* at 180°, but still better results the matter of balancing are obtained by the use of three or four cylinders with the cranks equally spaced about the crank-circle.

Among the automobile builders, the jump-spark is finding favor, as it may be better controlled at the high rotative speeds at which these engines run. The change of lead for the spark may be obtained by rotating the cam or switch, as the case may be, around the camshaft. Governors for automobile engines are very well in theory, to prevent the engine from speeding up when an obstruction throws one of the wheels off the ground, but there is no governor within the knowledge of the writer that will act within a time sufficiently short for the purpose. The speed of the engine should always be at the control of the operator. If the engine speed is under control, there will be but two changes of speed required, one for levels and easy grades and the other for hill-climbing. The reversing gear may be connected to the hill-climber as a high speed is undesirable for going backward.

For cooling the cylinders, both a closed circulating water system is employed, and also ribs radiating from the cylinder-walls. *The latter may* not be used successfully on *cylinders over* 3½ to 4 inches diameter. *For an* engine employing a water-jacket there should be some method of coolin the water, otherwise the amount of wate which it is necessary to carry become excessive. In France, where the gasolin automobile is in extensive use, a coil o pipe with collars to increase the radiatin surface of the pipe is considerably employe These collars or disks are stamped fro sheet metal and punched with a star-shape hole in the center and then forced over th tube, while the tube is straight. Afterwar the tube is coiled on a comparatively smal radius. This coil is made a part of th circulating system and answers its purpos very effectively. The author would sugges that if means were taken to guide ai through these coils by means of a curve shield under the carriage it would increas the cooling efficiency of the coil.

The gasoline and the water-tanks shoul each have some means by which the quan tity of the liquid contained in them coul be determined at any time without it bein necessary for the operator to leave his sea All parts of the mechanism should be co pletely enclosed to protect them from du For this reason an enclosed crank-cham is a desirable feature. Lubrication of ev part should be strictly automatic, so t when the carriage has been given proper attention at the outset of a

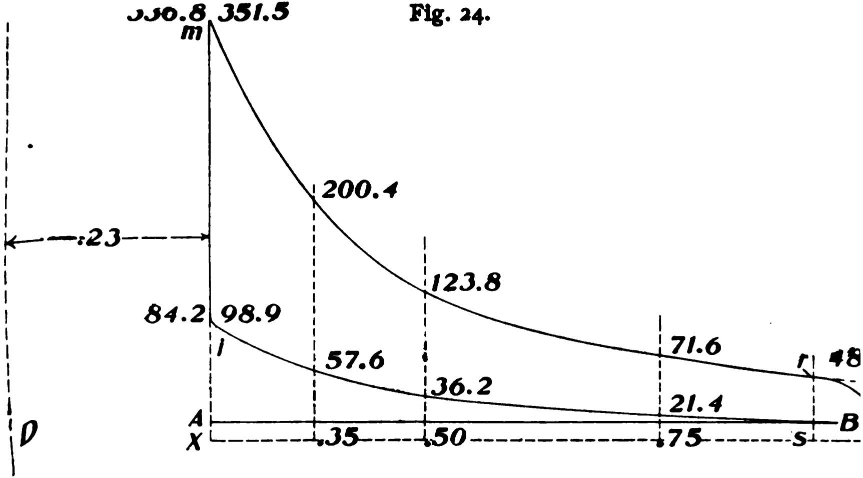


Fig. 24.

350.8
351.5
m
200.4
123.8
84.2
98.9
l
57.6
36.2
71.6
21.4
r
48
.23
A
B
X
.35
.50
.75
S
D

necessary to decide upon, either the compression pressure desired, or upon the ratio of the volume of the compression space to that displaced by the piston during the stroke. For the diagram in the figure, the ratio has been taken as 30 percent. The total volume of the cylinder, when the piston is at the end of its outward stroke, is therefore $1 + .3 = 1.3$ of the piston displacement. As it is much more convenient to consider the total cylinder volume as unity for the purpose of making the calculations, it is necessary to find what proportion of the total cylinder volume is included in the compression space, or to divide .3 by 1.3, giving .2308 as this ratio. In calculating the pressures at various points upon the curves, the following formulas should be used, because they represent the average behavior of the gases shown by actual indicator diagrams taken from engines in operation. For the compression curve

$$PV^{1.3}=K \quad (1)$$

Wherein P = the pressure above a vacuum or absolute pressure;

V = the volume of the gases the time they are at the pressure P;

K = a number or constant d *pending* upon the conditions, but whi *is the* same for all parts of the curve.

When the volume of the cylinder is considered as unity, K becomes the absolute pressure of the atmosphere, usually taken as 14.7. When natural gas is employed at the proportion to air which gives the best effect, the pressure of the gases after explosion is four times the pressure after compression, both these pressures being those above the atmosphere.

The formula for the expansion curve is

$$PV^{1.35} = C \quad (2)$$

Wherein P and V have the same significance as in formula (1) and C is a constant depending upon the maximum pressure.

These equations may also be written as follows:

$$PV^{1.3} = PV^{1.3}, \quad PV^{1.35} = PV^{1.35},$$

a form which shows the exact relation between any two points on the curve.

In order that the reader may better see how to apply the above formulas, the computation of the diagram in Fig. 24 will be given. Taking first the computation of the pressure at the end of the compression stroke, and applying formula (1)

$$PV^{1.3} = K = 14.7, \text{ and } P = \frac{14.7}{V^{1.3}} = \frac{14.7}{(2.308)^{1.3}}$$

In order to find the denominator of this fraction, it is necessary to multiply the true logarithm of the number by 1.3. The logarithm of .2308 as found in a table of logarithms (the tabular logarithm) is $\bar{1}.363236$, and since the logarithm has a negative characteristic (number to left of decimal point), the true logarithm must be found by adding the mantissa (number to right of decimal point) to the characteristic as follows:

$$\begin{array}{r}
1.000000 \\
\underline{.363236} \\
.636764 \\
\underline{1.3} \\
1910292 \\
\underline{636764} \\
\underline{.8277932}
\end{array}$$

$\bar{1}.172207$ (subtracting from 1 and adding —1 to get the tabular log).

Log 14.7 = 1.167317, and subtracting the log of $(.2308)^{1.3}$ the resulting log is that of the compression pressure at the end of the stroke.

$$\begin{array}{r}
1.167317 \\
\underline{\bar{1}.172207}
\end{array}$$

1.995110 = log of 98.88. 98.88 lb. is the compression pressure in lb. per sq. in. above a vacuum. For practical purposes, three figures will be sufficient and 98.9 may be taken as the pressure. The pressur above the atmosphere would be 98.9—1 = 84.2 lb. per sq. in. To get the maxi pressure, this amount should be multiplied by 4, giving $84.2 \times 4 = 336.8$ lb., and $336.8 + 14.7 = 351.5$ lb. absolute pressure.

In order that the proper shape of the curve may be very closely approximated, it is best to compute the pressures at three points between the ends. The points for which the calculations in the figure were made are those when the volumes are .35, .5 and .75, and in case of any uncertainty a point should be taken midway between .35 and the higher end of the curve. The pressures corresponding to the points given, are 57.6, 36.2 and 21.4 respectively.

The pressures for the expansion curve are found in the same manner as for the compression curve, but by means of formula (2). Before this formula is applied, however, it is necessary to find the value of the constant C, which is the pressure at the end of the stroke when the volume is equal to 1. Hence $PV^{1.35} = C$ becomes $351.5 \times (.2308)^{1.35} = C$.

Log $(.2308)^{1.35} = \bar{1}.140368$, Log $351.5 = 2.545925$. Adding these logs gives, for the sum, the log of the constant C, thus:

$$\begin{array}{l} \bar{1}.140368 \\ \underline{2.545925} \\ 1.686293 \end{array} = \log 48.56.$$

48.56 lbs. is the terminal pressure of the gases in the cylinder, *should release* take place at the end of the stroke. Hence the equation of the expansion curve is $PV^{1.35} = 48.56$.

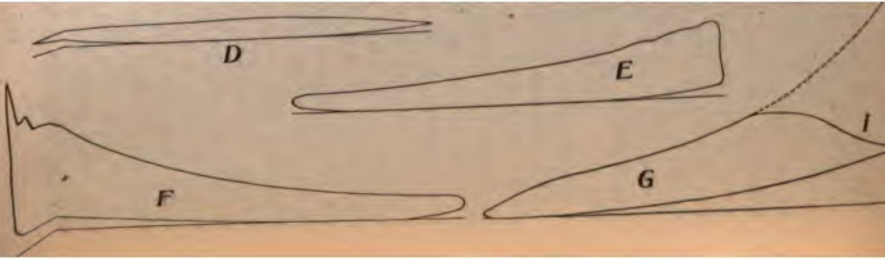
The intermediate pressures for the remainder of the expansion curve are found by subtracting the logs of .35, .5 and .75 raised to the 1.35 power, from the log of 48.56. They are 200.4, 123.8 and 71.6 respectively. After these points are located, the diagram is constructed by drawing the curves through them as shown in the figure. The length of the diagram is 3½″ from the line *D C* to *e*, and the scale of the spring represented is 160 lb. Hence unit volume is represented by a length of 3½″, and a height of one inch on the diagram represents a pressure of 160 lbs. per sq. in.

In order to make the diagram complete, it is necessary to add the two small curves at *i* and *r e*. The curve at *i* is that due to the lead of the ignition which causes an increase of the rate at which the pressure is rising. It can be drawn but approximately, as it varies in size for any change in the piston speed. Its effect upon the area of the diagram is too small to be of consequence, and the only value it has, is to serve as a memorandum when using the diagram for the purpose of comparison with those actually taken from the engine. The curve *r e* is that produced by the release taking place before the end of the stroke is reached

The point of release *r* should be such as to bring the point *e* where the expansion line meets the atmospheric line *A B* at the end of the stroke. To draw this curve as it is shown in the figure, erect the vertical line *s r* at a distance from *Xm* equal to .9 time the length of the diagram, which, in the figure, is .9 × .77 = .693 time the unit length. Through *r* and *e* describe a circular arc, which shall be tangent to the expansion line at *r*. Joining the points *i* and *m* with a straight line completes the diagram.

The diagram having been drawn, the designer may find the M. E. P. by means of planimeter as explained in the chapter on Testing. From the M. E. P. obtained from this diagram, he may calculate the horse-power of an engine of any size which it is proposed to build, or he may determine the dimensions of the engine. The design of the valve motions may, to a certain extent, be founded upon the diagram, as it shows the proper time for opening and closing the valves. It should be the aim of the designer to build an engine that will give as nearly as possible, such a diagram as that shown in the figure, for only with a diagram of this shape are the best results obtained.

A few diagrams will now be shown which *have been made* by an engine when not *working under* the conditions necessary for


D
E
F
G
I

producing a diagram similar to that in Fig. 24. In Fig. 25 are shown diagrams taken from various engines while in actual operation and they illustrate just what occurs under several conditions. At (A) is shown a diagram taken from an engine which is operating with everything in good working order with the exception of an exhaust release which is a trifle late, indicated by the expansion line not reaching the atmospheric line *X Y* until the piston has returned to *d*. The point of release is shown by the sudden change in the curvature of the expansion line at *c*, the point of ignition by the sudden change of curvature in the compression curve at *i*.

On either side of the atmospheric line is a curve which is exaggerated in the drawing in order that it may be distinct. The upper curve is that produced by pressure within the engine cylinder on the exhaust stroke, and is due to an unnecessarily contracted exhaust passage, or to obstructions that have accumulated in the exhaust-pipe. The lower curve is caused by the pressure within the cylinder falling below that of the atmosphere, forming a partial vacuum because of an obstructed inlet passage. Ordinarily, these curves are too small to be of any moment, but in case they are a quite distinct departure from the atmospheric line, they show that an unnecessary amount of work is being done by the piston in forcing the gases through the passage, and steps should be taken to remedy the defect. A condition of this kind is spoken of as wire-drawing. Diagram (B) is that produced when the ignition of the charge is late. In this case ignition takes place at *i* after the crank has passed the center. The loss in area of the diagram is quite noticeable when comparing (B) with (A).

The reader should learn to distinguish between a diagram produced by late ignition and that produced by a weak or a throttled mixture. In (B) the curve indicating rise of pressure immediately after ignition, is concave, while in (D) and (E), which are diagrams produced by a weak mixture, this curve is either a straight line or is convex. (G) is a diagram in which the ignition is very tardy, and the area lost because of late ignition is shown by the dotted lines. Premature ignition is shown by (C). In this diagram the ignition is at *i* and the loss of area is illustrated by the proper diagram being shown in dotted lines. Diagram (F) is an example of the effect of late release together with obstructed exhaust passages.

When an engine is operating on fuel which ignites at a low temperature, as is

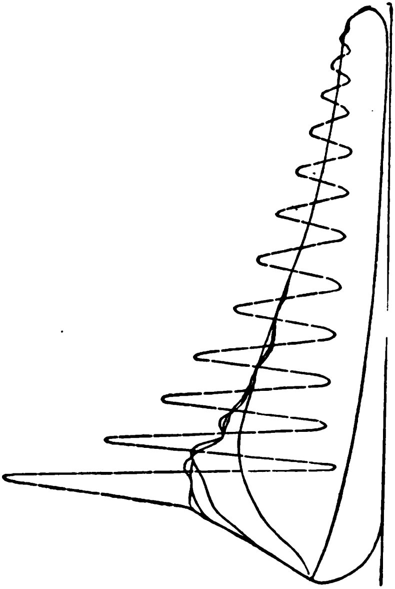


Fig. 26.

the case with gasoline and acetylene, the propagation of the flame throughout the mixture is very rapid, causing an explosion which is almost instantaneous. An example of the sort of diagram usually obtained with these fuels is shown in Fig. 26. The sudden blow given to the piston of the indicator, sets the spring in vibration and produces the wavy line which appears at the point immediately following maximum pressure. This effect has been attributed to various causes, such as a rapid succession of explosions, and these in turn have been said to be caused by an uneven distribution of the fuel within the air (a stratified charge). The author believes that very few of these curves are of such a nature as to justify such a reasoning. They are all of the nature of a sine curve, the waves gradually lessening in height as the piston proceeds on its stroke. In using these fuels, it is best to make the lead of the igniter somewhat less than when using natural gas or manufactured gas. By *lead* is meant the time or the distance before the end of the stroke, that ignition takes place.

CHAPTER XV.

GAS-ENGINE DIMENSIONS.

The power of any engine is dependent upon four factors, the average or mean effective pressure upon the piston during the stroke, the area of the piston upon which this pressure is exerted, the length of the stroke, and the number of times per minute the pressure is exerted. The power of an engine computed from the above factors is known as the indicated horsepower (I. H. P.). The power delivered at the pulley or obtained from the engine, and which is available for power purposes, is called the delivered horsepower (D. H. P.) or the brake horsepower (B. H. P.) and by some writers the effective horsepower (E. H. P.). The use of the latter abbreviation is, however, likely to cause confusion as the same abbreviation is employed for electrical horsepower. The term brake horsepower is derived from the manner of testing the outpu of the engine, which is usually with some form of brake. The D. H. P. is the I. H. P. minus the power absorbed in the friction of the engine, the latter being called the friction load. The ratio of the D. H. P. to the I. H. P. is called the mechanical efficiency of the engine, sometimes abbreviated to M. E., and M. E. $= \frac{\text{D. H. P.}}{\text{I. H. P.}}$, this ratio being usually expressed in percent. Thus, if the I. H. P. of an engine is 10 H. P. and the D. H. P. is 8 H. P., the M. E. $= \frac{8}{10} = 80$ percent, and the friction load $= 10 - 8 = 2$ horsepower.

The mean effective pressure (M. E. P.) in a gas-engine cylinder varies with the fuel, the pressure after compression, the proportion of the mixture and the time of ignition, and in a minor way on several other conditions which are not of sufficient importance to consider at present. If the mixture contains the most effective proportion of gas and air, and the ignition is properly timed, as it should be, it leaves the matter of compression and quality of fuel as the two important points to be considered, with reference to the M. E. P. The highest mean effective pressures are those obtained with natural gas and gasoline, then come the illuminating gases in the order of their light-giving values—water-gas and semi-water-gas—the latter being usually known as producer-gas. For the compression pressure usually employed, the M. E. P.s range between 45 lb. per sq. in. and 80 lb. per sq. in., with the average not far from 65 lb. per sq. in. for an average quality of natural gas. For gasoline the M. E. P. may safely be taken at 70 lb. per sq. in. The author has found the following formulas are very well borne out in practice for the best performance of the average gas engine:

Let D = the diameter of the cylinder in inches;

L = the length of the stroke in inches;

R = the number of revolutions per minute;

Then for a four-cycle engine

$$\text{D. H. P.} = \frac{D^2 \times L \times R}{19{,}000} \qquad (3)$$

For a two-cycle engine

$$\text{D. H. P.} = \frac{D^2 \times L \times R}{14{,}000} \qquad (4)$$

For engines using gasoline these denominators should be reduced to 18,000 and 13,500 respectively. These formulas may be used for determining the dimensions of any engine that is being designed, if the performance of any engine of the same kind and on the same quality of fuel is *already known*. The denominator of the [illegible] then becomes an unknown quantity and it may be found as follows:

Calling the unknown denominator X

$$X = \frac{[illegible]\ R}{D.\ H.\ P.} \qquad (5)$$

Suppose an engine which is a good example of a series already being built has a cylinder [illegible] diameter and a stroke of 30″, [illegible] engine, and that it runs at [illegible] D. H. P., the value of X is

$$X = \frac{[illegible]}{[illegible]} = 20,577 +$$

[illegible]

The formula for these engines is then

$$D.\ H.\ P. = \frac{[illegible]\ L \times R}{20,600}$$

To find the diameter of the cylinder for any engine to give a required horsepower at a given speed, the ratio of the stroke to the diameter should first be determined. For a four-cycle engine, prominent authorities agree that the best proportion is to make the stroke 1½ times the cylinder diameter. For a two-cycle engine, practice varies between making the diameter of the cylinder and the stroke the same, to a stroke equal to 1¼ times the diameter. It is al necessary to decide upon the spe which the engine shall run. Gas built in the United States, are at a higher piston speed for horizontal engines than 600 feet per minute and for vertical engines 700 feet per minute. The following formulas representing average practice among manufacturers will be found valuable in making the first approximate calculation:

Let $H =$ the D. H. P. of the engine;
Let $R =$ the rev. per min.;

Then for a four-cycle engine $R = \dfrac{380}{(H)^{.21}}$ (6)

" " two-cycle " $R = \dfrac{440}{(H)^{.21}}$ (7)

In order to solve the above two equations it is necessary to use logarithms. Suppose it is desired to find the speed of a fifteen-horsepower four-cycle engine. Take a table of logarithms and find first the logarithm of 15, which is 1.176091; multiplying by .21 the result is .24697911, which is the log of 15 to the .21 power. The log of 380 is 2.579784; subtracting the log of $(15)^{.21}$ from this—

$$\begin{array}{r} 2.579784 \\ \underline{.246979} \\ 2.332705 \end{array} = \log 215.1 +$$

And the proper speed for this engine would be 215 r. p. m. or thereabout. For a four-cycle engine, formula (6) gives a piston speed of 600 feet per minute at 32.5 horsepower, and for a two-cycle engine, formula (7) gives a piston speed of 600 feet per minute at 71 horsepower. Beyond these powers, the formulas (6) and (7) should not be used but the computations made from the piston speed. At 600 feet piston speed the r. p. m. is found as follows:

$$R = \frac{3600}{L} \qquad (8)$$

When the piston speed of the engine is 600 feet per minute, much simpler formulas than (3) and (4) may be used to find the diameter of the cylinder. Using the same value of X as before,

$D = 2.3 \sqrt{H}$ for a 4-cycle engine; $X = 19{,}000.$ (9)

$D = 1.67 \sqrt{H}$ for a 2-cycle engine; $X = 10{,}000.$ (10)

These formulas should not be used except for a piston of speed of 600 feet per minute. The stroke of the engine can afterwards be determined by its ratio to the diameter, and from that the r. p. m. may be found by means of formula (8). It should be remembered that these equations are applicable as they stand, only for engines with one cylinder. When the engine is to be built with two or more cylinders, the horsepower must be divided by the number of cylinders. Thus, if a three-cylinder engine of 45 horsepower is to be designed, the calculations for cylinder diameter, r. p. m., etc., should be made for a 15-horsepower engine.

Should the reader desire to calculate the power of a gas engine from a known M. E. P., the following formulas will assist him. When the piston speed is 600 feet per minute and the stroke of the piston is 1½ times its diameter

$$D = 1.67\sqrt{\frac{\text{I. H. P.}}{P}} \qquad (11)$$

Wherein P = the M. E. P.

When the piston speed is 700 feet per minute

$$D = 1.55\sqrt{\frac{\text{I. H. P.}}{P}} \qquad (12)$$

Both of these formulas are for the 4-cycle engine. If it is desired to find the D. H. P., an M. E. of 80% may be safely employed for engines under 25 horsepower. For larger engines having two or more cylinders 85% may be used. The reason that a higher efficiency may be obtained from multiple cylinder engines, is that the flywheels are much lighter, and the load is better distributed throughout a revolution, reducing the friction of the engine. It is fast becoming the custom to build engines *larger than* 50 D. H. P. with two or more

dimensions of the engine an
them vary but a few percen
would be given by the best
Although every dimension of
should rightly be discussed in
chapter, the author will give
chapter by itself in order to
easier of reference.

CHAPTER XVI.

THE CYLINDER.

A good example of a gas-engine cylinder is shown in Fig. 27. The dimensions of the various parts have been based upon average practice of gas-engine designers in this country, and, although he will not be able to take up any gas-engine design he chances to come across and find the dimensions exactly fitting these formulas, he will, by the use of the following equations, be able to produce a design which will be in good proportion.

For the thickness of the cylinder wall it is necessary to allow for reboring to a certain extent, and also to have sufficient metal for stiffness, and to make a good casting when the cylinder is a small one. The thickness of the wall is therefore made quite a little heavier than is necessary for mere resistance to pressure. The following formula is that which represents average *practice.*

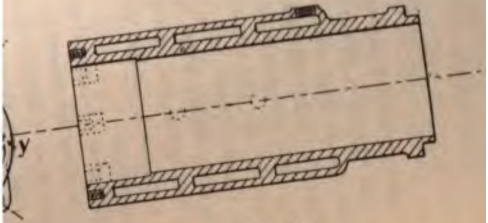


Fig. 27.

$$t = .09\,D; \qquad (13)$$

Wherein t = the thickness of the cylinder wall;

D = the diameter of the cylinder.

The depth of the water-jacket is usually as given by the following formula:

$$j = .1\,D; \qquad (14)$$

Wherein j = the depth of the water-jacket. This depth is measured on a radius of the cylinder across the water-jacket.

No regular proportion appears to be followed at all closely for the thickness of the outer wall of the water-jacket. The author has used for engines of his own design a thickness which is half that of the cylinder wall, or .045 D.

For very small engines these formulas give dimensions which produce walls too thin for the foundryman. In order to get good castings it is well to limit the thickness of the cylinder wall to $\frac{5}{16}$ inch, the depth of the water-jacket to $\frac{3}{8}$ inch, and the thickness of the jacket wall to $\frac{1}{4}$ inch.

In order to find the diameter of the cylinder head studs the maximum pressure within the cylinder must be known as well as the number of studs. It is a good plan to limit the distance between the studs to six inches, unless there is a feature in the design of the cylinder or the head that will prevent such a space being used. An even number of studs is almost invariably employed. The following formula should be used for finding the diameter of the stud:

Let A = the diameter of the cylinder;
z = the diameter of the stud at the root of the thread;
S = the safe stress in lb. per sq. in.;
p = the maximum pressure in the cylinder;
n = the number of studs;

Then $z^2 = \frac{D^2p}{sn}$ or $z = D\sqrt{\frac{p}{sn}}$. (15)

The flange by which the cylinder is attached to the frame, may be either square or circular. It is usually square on vertical engines and circular when the cylinder is horizontal. The sizes of the bolts which are used to fasten the cylinder to the frame may be found by formula (15) making z the diameter of the bolt at the root of the thread and n the number of bolts. The thickness of the cylinder wall between the end of the water-jacket and the flange should be .125 D. The thickness of the flange should be equal to the diameter of the bolt + ⅛″. On engines smaller than six horsepower, it is customary to make the cylinder and the frame in one casting. The water-jacket should extend, beyond the end of the piston nearest the cylinder head, to

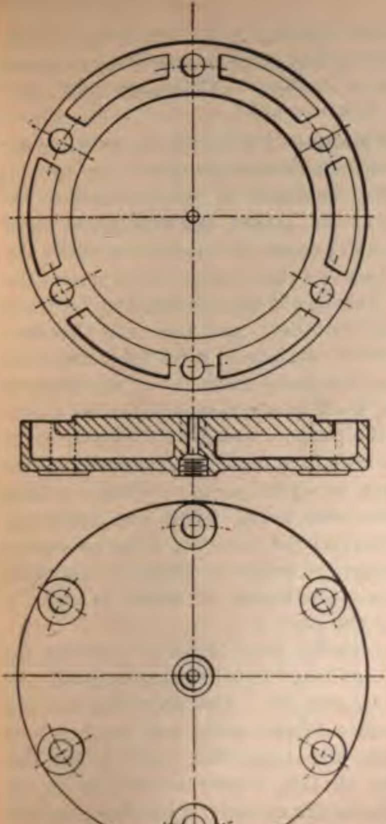


Fig. 28.

a distance equal to about 10% of the stroke. The size of the water inlet and outlet pipes should be .15 times the diameter of the cylinder.

Two methods are in use for making the core for the water-jacket space. In one the core-box consists of two cylinders of wrought iron, placed one within the other and of the proper dimensions to make the space between them equal to the size of the core. The core is then rammed up between the two cylinders and put into the core-oven in the core-box. With this method of making the cores there is no necessity of making draft in the core-box for the metal about the studs. Another method of making the core is to make one half of the core at a time, using the same core-box for each half, the core being lifted out sideways. With this form of core-box it is necessary to arrange the metal as shown by the dotted lines in the figure in order to make a draft on the parts *y y*.

The cylinder head, when it contains no valves or other mechanism, is made as shown in Fig. 28. The formulas for the thicknesses of the walls and the depth of the water-jacket are the same as for the cylinder. In fact, it may be said to be an extension of the cylinder. In some engines the end of the cylinder is closed and is made in the form of a sphere. In others the cylinder is closed but a head is used to cover the end of the cylinder in such a manner that it forms the outer wall of the water-jacket at that point. In many engines, the valves are placed in the head while in others, the head contains but one valve or the igniter.

CHAPT

VALVES A

The methods
charge and the cy
Chapter X. The
passages, the valv
will be discussed
The proportions
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be founded upon
gases will be
them. Hence th
depend upon both
and the speed of
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per second

per second in the exhaust passages. The 85 feet per second allowance for the exhaust, is made on the assumption that the products of combustion are driven from the cylinder when at atmospheric pressure, this assumption being made for convenience only. As a matter of fact the pressure of the gases at the moment of release range between 30 and 40 lb., and the lower limit of speed for the gases in the exhaust passages is adopted for this very reason. As the piston speed of gas engines varies considerably, even for engines of the same power when built by different manufacturers, it should always be taken into consideration when proportioning the passages. It should also be remembered that to make the areas of the inlet and the exhaust pipes according to the above, and to then choke the column of gas by a small valve opening or cylinder port, is not very good practice. The formulas given below will give the designer a short method of determining the sizes of the passages:

Let $S =$ the speed of the piston in feet per minute;
$A =$ the area of the cylinder;
$a =$ the area of the inlet passages;
$a^1 =$ the area of the exhaust passages;
$D =$ the diameter of the cylinder;
$d =$ the diameter of the inlet passage;

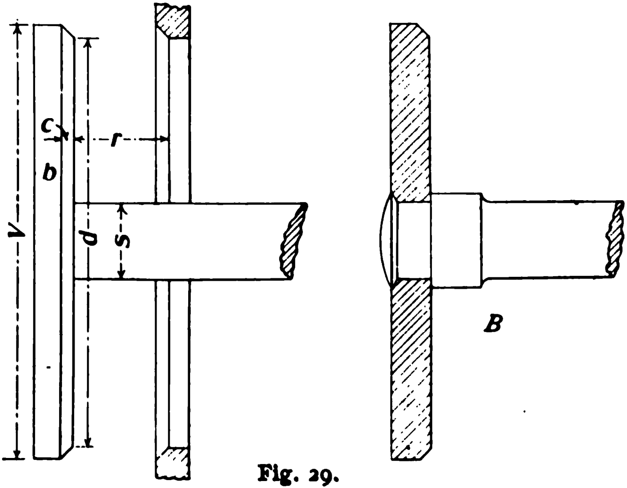


Fig. 29.

d^1 = the diameter of the exhaust passage;

L the length of the stroke in inches;

R = the r. p. m. of the crankshaft;

then

$a = \frac{AS}{6{,}000}$ (16) or $a = \frac{ARL}{36{,}000}$ (16a) for the inlet.

$a^1 = \frac{AS}{5{,}100}$ (17) or $a^1 = \frac{ARL}{30{,}600}$ (17a) for the exhaust.

$d = .00515\,D\sqrt{RL}$ for the inlet (16b).

$d^1 = .00572\,D\sqrt{RL}$ for the exhaust (17b)

When the piston speed of the engine is at the usual limit for horizontal engines these formulas may be still further simplified to:

$a \quad .1\,A$ (16c);

$a^1 = .12\,A$ (17c);

and for passages of circular cross-section to

$d = .316\,D$ (16d);

$d^1 = .35$ D (17d).

The designer should use the formula best suited to his work and he will find that, in many cases, he may calculate the dimensions by one formula and use another as a check.

The proportions of the valve are shown in Fig. 29. The author does not insist that these proportions should be followed in *every case* but they form a very convenient

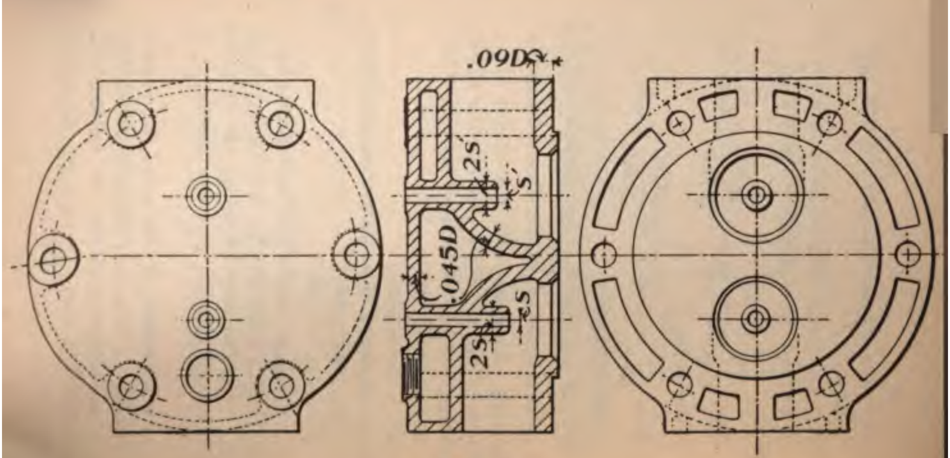


Fig. 30

basis for the design of a valve, especially in those cases where it is hard for the designer to decide what to do, as the ratios suggested below will give a valve of good proportion. The dimensions are based upon the diameter of the opening as a unit. They are:

$V = 1.14\,d$;

$b = .14\,d$;

$c = .07\,d$;

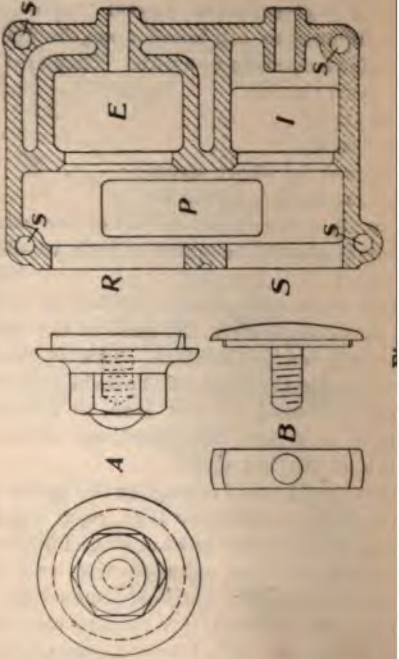
$s = .2\,d$, for short stems;

$s = .23\,d$, for long stems;

$r = .25\,d$, the minimum lift of the valve, to which should be added $\frac{1}{8}$ inch to allow for wear.

At *B* is shown the usual method of making an exhaust valve, the head of the valve being riveted to the stem. This is necessary because the head of the exhaust-valve should be made of cast iron, as wrought iron or steel will not stand the abrasive action of the hot gases.

Three methods of arranging the valves on the engine are shown in Figs. 30, 31 and 32. Fig. 30 is a cylinder head arranged for both the inlet and the exhaust-valve, and the proportions of the various dimensions to the cylinder diameter are marked upon the different parts. It is always a good practice to water-jacket the exhaust-valve, so that the heat may be carried away from both the bearing for the stem and the valve-seat.


S
S
S
S
E
I
P
R
S
A
B

In the figure, the inlet-valve is also shown water-jacketed. It is not necessary to jacket the inlet-valve, as the incoming air suffices to keep it cool. The jacket is continued about the valve in this case, in order to save iron and to retain a uniformity of construction.

In Fig. 31 is shown a valve-box designed to be attached to the side of the cylinder and to so place both valves that their stems are parallel and point in the same direction. This box is a style used on both horizontal and vertical engines and is employed where the camshaft is at a right angle to the axis of the cylinder and between the valve-box and the crankshaft. In order that the valves may be removed from the box without taking them apart, openings are made in the box at *R* and *S* and the holes are closed with a cap *A*. The cap is fitted to the box with a ground joint and is held in place by the clamp *B*. The proportions of the various parts of the box are shown by formulas on the figure. But one view of the box is shown, a cross-section through the axes of the valves. It is bolted to the cylinder by means of studs passing through the holes *s*, the studs serving also to hold the cover of the box. The inlet gases enter through a cored passage in the cylinder and *the opening I*, from whence they pass

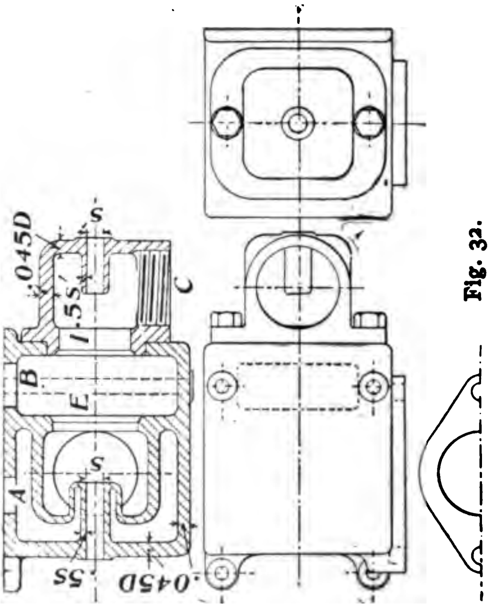


Fig. 32.

through the inlet-valve to the cylinder-port *P*. From the port *P* the products of combustion pass through the exhaust-valve opening to *E* and through to a cored exhaust passage in the cylinder.

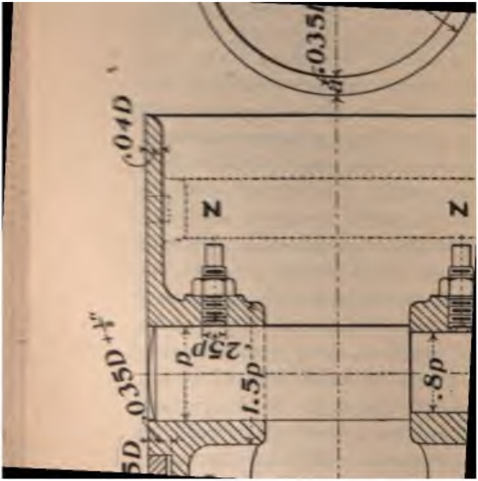
Fig. 32 is an illustration of a valve-box so arranged us to place both valves in a box of the smallest possible compass. The exhaust-valve is operated by means of a cam, but the inlet-valve is intended to be operated by the suction of the engine piston. The box is shown in section through the plan. In order to permit of removal of the valves, the seat for the inlet-valve and the bearing for its stem are placed in a separate casting, which is bolted to the box as shown. The opening from the valve-box into the cylinder is shown at *B* and from the jacket surrounding the exhaust-valve to the jacket of the cylinder at *A*. The exhaust-valve opening is shown at *E* and the inlet-valve opening at *I*. The exhaust-valve is bolted to the box by means of a flange, while the inlet is screwed into the side of the cover at *C*. It would be well to put a union into the inlet pipe near the box, in order that the cover may be taken off without difficulty. The dimensions of the the various parts of the box are shown by equations on the figure, as in Figs. 30 and 31.

CHAPTER XVIII.

THE PISTON, THE CONNECTING-ROD AND THE CRANKSHAFT.

With few exceptions, all gas engines are fitted with trunk-pistons. The principal reason for the adoption of the trunk-piston is, that if the engine were made double-acting, as are most of the steam engines, the hot gases would surround the piston-rod heating it to a temperature that would cause it to cut in the stuffing-box. The trunk piston taking an impulse on the end where there would be no rod were the engine double-acting, gives no trouble from this source. As the other end of the cylinder is not used, there is no necessity for closing it, and by making the piston long enough to act as a guide, there is no need for either a piston-rod or a cross-head. A few engines are manufactured which are double-acting, the trouble from an overheated piston-rod being avoided by providing means for cooling the rod. In one of these engines the *water-jacket* is extended so that it surrounds the stuffing-box, while in another, the rod is made hollow and water is circulated through it.

The present discussion will be confined entirely to the trunk-piston. For a treatment of the piston for the double-acting engine, the reader is referred to works on steam-engine design. An example of the trunk-piston, as usually employed with the gas engine, is shown in Fig. 33. The proportions are, with the exception of the pin, based upon the cylinder diameter as a unit. All the necessary equations are placed upon the drawing. For small pistons of 8″ diameter and under, the ribs *R* may be omitted, and the piston designed as shown by the dotted line. In some engines there is a supplementary piston-ring added, as shown by the dotted lines at *Z*. Up to and including pistons 10″ diameter three rings will be found sufficient. For sizes above 8″, four rings should be used, one of which may be placed as shown at *Z*. In pistons above 12″ diameter four rings should be placed between the pin and the inner end, and a fifth ring added at *Z* if it is desired to place a ring at this point. The extra ring is of somewhat doubtful utility, so far as the packing ring *Z* is concerned. It is contended by some that it relieves the piston of pressure on the cylinder, but as the piston


.035
.04D
N
N
.035D+1/8"
5D
p
.25p
1.5p
.8p

is not in any way supported by the rings, it is difficult to see what foundation there is for such an argument. In using the equations given in Fig. 33, the designer should be careful to observe the limits, on the smaller sizes, that it is necessary to consider for securing good castings. The piston wall should not be less than $\frac{3}{16}''$, and even then the pattern should allow considerable stock to be turned out in the lathe. The difference in the thickness of the rings at the smaller and the larger ends may be less or more than $\frac{1}{16}''$ according to the method of turning them up. A ring which would be of equal strength throughout would taper off to nothing at the smaller end. In the shop when turning piston-rings for a gas engine, some method should be followed whereby the outside of the ring is turned to a perfect circle, the diameter of which shall be exactly the same as the bore of the cylinder. In order to do this it is necessary to first cut the rings and then to clamp them in a jig which will hold them sprung very nearly together. The ring is then turned to the diameter of the cylinder while held in the jig. This is the secret of securing a tight piston, and one that will successfully withstand the high pressures due to the increase in compression, which is now so extensively coming into use.

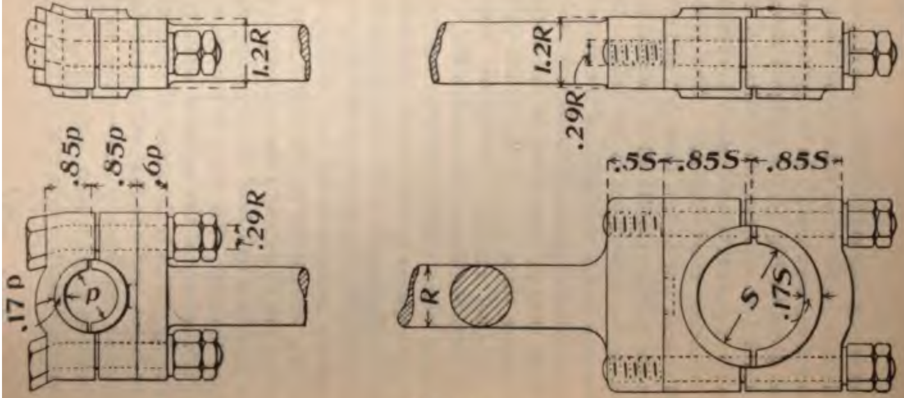


Fig. 34.

The diameter p of the piston-pin should be determined by the bearing surface required; the length of the pin should be, wherever possible, the same as that of the crank-pin. The average pressure which is allowable upon the piston-pin per sq. in. of projected area is 750 lbs. In order to find the diameter of the pin, when the length is known, the designer may make use of the following equation.

$$p = \frac{A\,P}{750 \times e} \qquad (18)$$

Wherein p = the diameter of the piston pin;
A = the area of the cylinder:
P = the M. E. P.;
e = the length of the piston pin.

The connecting-rod of a gas engine is made in various styles, but what is known as the marine type, and illustrated in Fig. 34, is that in most general use. This rod is of circular cross-section and is easily machined, as it is nearly all of it turned in the lathe. It is made slightly tapering, the larger end being that nearest the crank-pin. The taper given is dependent to some extent upon the length of the rod, the crank end being from ¼″ to ¾″ larger than the piston end. Thus, for a 24″ rod the difference would be ¼″, and on a 60″ rod a difference of ⅝″, or a difference of ⅛″ for

These formulas are for the average case of $L = 1\frac{1}{2} D$. When there is much of a departure from this ratio the following formula should be used:

For wrought iron,

$$S = .056 \sqrt[3]{m L D^2} \qquad (20\ b)$$

For steel, $S = .052 \sqrt[3]{m L D^2} \qquad (20\ c)$

These formulas will give somewhat larger crankshafts than are in use on many gas engines, but it is a matter worthy of note that too many crankshafts are made too small. The diameter of the crankpin is usually made 1⅛ times that of the crankshaft. Practice varies in this respect from making the pin the same diameter as the shaft to 1.25 times the shaft. It is a good plan to select the larger rather than the smaller ratio. Mr. Frederick Grover, an English writer, recommends the ratio 1.2 as the lowest limit. It should be remembered that the larger the diameter the shorter the pin. The projected area of the crankpin should be such that the average pressure does not exceed 400 lb. to the sq. in. From this the following formula is derived:

$$f = \frac{A\ P}{400\ q} \qquad (21)$$

Wherein q = the diameter of the pin;
f = the length of the pin;
A and P as in formula ([illegible]

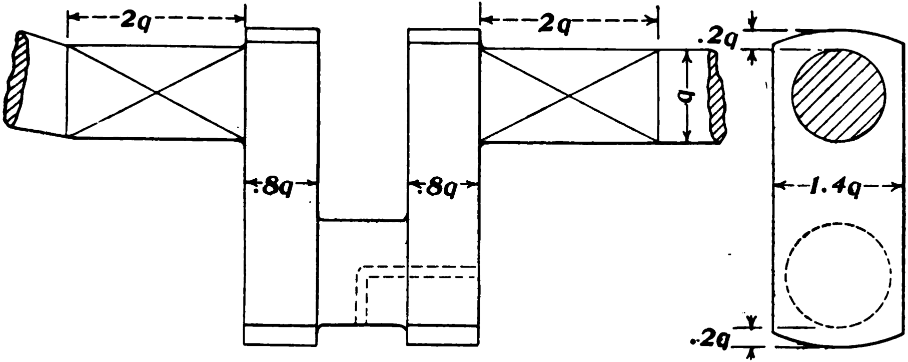


Fig. 35.

The length of the crankshaft bearing usually from $2S$ to $2.5S$, the smaller ra $2S$ being that in most general use. Fig shows a good example of a crankshaft w those proportions not already given, mark upon the figure.

CHAPTER XIX.

THE ENGINE FRAME.

As with many other parts of the gas engine, the frame is built in so many different styles that to give an example of every one would take up more room than is at the disposal of the author in this work.

A good example of the style of frame in use on a great many horizontal gas engines is shown in Fig. 36, with the proportions of the various parts shown by their relation to the cylinder diameter of the engine. The angle of the parting line of the bearing brasses is taken at 45°, in order that the thrust of the piston may not come directly against the stud, but against the frame. The most rational angle for the parting line would be at 90° to the resultant of the average total pressure of the piston and that of the flywheels. This would, however, make the angle a very steep one, and it makes a much neater-looking design to make the angle as shown, while keeping

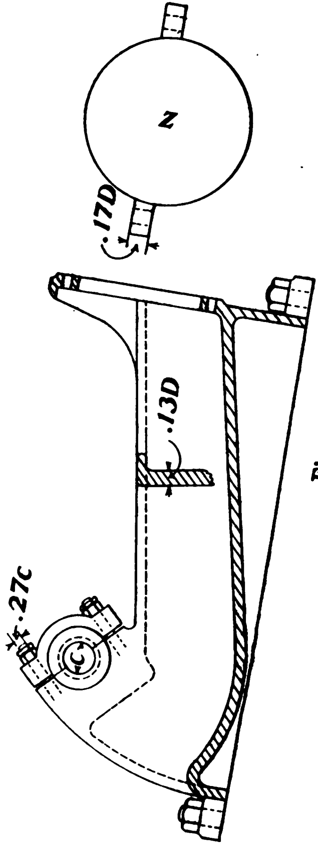


Fig. 36.

the pressure of the thrust from the piston, on the frame. The cylinders of gas engines are quite frequently cast with projections on the side, by which they are bolted to the frame, as shown in cross-section at *Z*. It is claimed that the overhanging piston will not get out of shape when the temperature rises while the engine is at work. The author must confess that he has been unable to detect any difference in the effect upon the cylinders with an engine built by either method. In some engines the frame is set directly on the foundation, while with others a sub-base is furnished, which is practically a continuation of the frame. It is usual to make the lower part of the frame hollow when there is no base, and to draw the air supply from this space in order to make the suction a quiet one. When there is a base, the air is drawn from the hollow portion of the base.

CHAPTER XX.

FLYWHEELS.

The great proportion of idle strokes to those in which work is actually being done upon the piston makes it imperative that the gas engine be supplied with a very heavy flywheel in order to regulate the speed within reasonable limits. For different requirements in speed regulation there will be a difference in the weight of the flywheel. Steadiness of speed is usually spoken of as percent variation, for the reason that there is no such thing as absolute uniformity of speed, but a variation between limits that is determined by the efficiency of the governor mechanism and the regulating power of the flywheel. Thus, a gas engine that is operating with a speed variation of 2% and at an average speed of 200 r. p. m., has a difference between the highest and the lowest speed of the engine occurring between one impulse stroke and the next, of 200×.02=4 revolutions; and since the average speed of the engine should be 200 r. p. m., the speed should vary between 202 r. p. m. and 198 r. p. m. It is as well to note, at this point in the discussion, a mistake that is occasionally made. This is, to consider the steadiness of speed as the percent of difference of speed between no load and full load. To illustrate: If the engine ran at 220 r. p. m. at no load, and at 212 r. p. m. at full load, then the speed variation would be considered as 220—212=8 r. p. m., and the engine said to regulate within 8÷216=3.7% nearly. A little reflection will show that the steadiness of motion is not to be determined by the difference between the speeds at no load and at full load, this matter being one that is dependent in a great measure upon the regulating power of the governing mechanism; or, in other words, upon the efficiency of the governor. The steadiness of speed between one impulse and the next is dependent entirely upon the power for storing energy that is contained not only in the flywheel of the engine but in the moving parts of the machinery it is driving. Thus, the armature of a dynamo adds somewhat to the power of storing energy, as do the pulleys, etc., of any pieces of machinery being *driven. In the* operation of electric generators for incandescent lighting, the regulating power of the engine flywheel is quite frequently augmented by the use of an additional flywheel on the armature shaft, on a jack-shaft between the engine and the dynamo, or on both the dynamo shaft and the jack-shaft.

It is obvious that the rational method of calculating the energy storing power required in the gas-engine flywheel is to make a graphical diagram of the operations that take place within the engine cylinder and to transform them into a diagram showing the resultant effect of these forces upon the crank-pin. There are two ways in which this may be done. One of these methods considers only the effect of the pressures within the cylinder, while the other and most accurate method brings in the effect of the reciprocating parts of the engine, *i. e.*, the piston, the connecting-rod and the cross-head when it is employed. Both of these methods require tedious calculations, and the reader is referred to works of greater scope than the present one for description and explanation of these methods. For the case of a gas engine operating under average conditions the following formulas will be found to give good results in practice. They give values that are within a very few percent of those obtained from the actual diagram of the engine, and show good results in practice:

Let W = the weight of the flywheel rim in pounds;
f = the diameter of the flywheel at the center of gravity of the rim in inches;
N = the r. p. m. of the crankshaft;
E = the coefficient of unsteadiness permissible;

$$\text{Then } W = \frac{\text{I. H. P.} \times 111{,}600{,}000{,}000}{f^2 N^3 E} \qquad (22)$$

when the engine has an impulse at every fourth stroke. If the I. H. P. is taken as the power of the engine when operating at less than the maximum number of explosions, the value of the figure in the numerator must be increased in proportion. Thus, should the calculations be made for an I. H. P., taken when the engine is getting an impulse only every eighth stroke, the weight found by the above formula must be multiplied by 2. The value of the coefficient E for the various classes of work is given by competent authorities to be:

For Pumping and ordinary duties	.05 ;
For driving machine tools	.03 ;
For driving textile machinery	.025;
For driving dynamos	.02 ;
For driving spinning machinery	.01 ;
Marine	15.
Automobiles	.335.

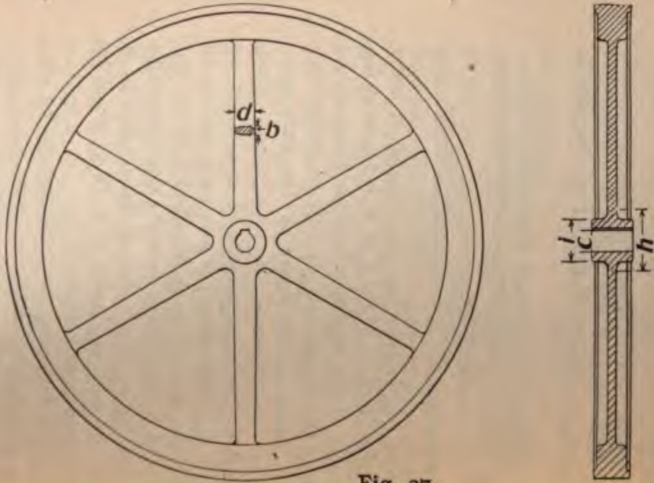


Fig. 37.

If the flywheel on the engine is to be assisted by flywheels on a jack-shaft, or on the armature shaft of a dynamo or other machinery it may be driving, the power storing effect should be divided among the various flywheels. This is done by first deciding what power is stored by the flywheels on the engine at the coefficient of steadiness employed, and then finding by means of formula (22) what weight of flywheel is required on the jack-shaft to store the remainder, using for f and N the diameter to the *center of gravity* of the jack-shaft flywheel and the r. p. m. of the jack-shaft.

The general proportions of a gas-engine flywheel are shown in Fig. 37. Taking the diameter of the crankshaft as a basis of the dimensions of the hub and the spokes, the proportions are as follows:

$i = 2c.$

$h = 3c.$

$d = .8c$ to $1.2c.$

$b = .4d$ to $.5d$ about. This dimension should, however, be calculated from the following formula:

$$b = \frac{B.\,H.\,P. \times 175}{nNd^2} \qquad (23)$$

Wherein $B.\,H.\,P.$ = the maximum brake horsepower of the engine;

n = the number of spokes;
N = the r. p. m. of the crankshaft.

It appears to be the universal practice to use six spokes for a gas-engine flywheel. The length of the hub should be at least $1.5c$, and is usually from 1.75 to 2.5 times the diameter of the crankshaft. The dimensions of the rim must be made to suit the requirements of the engine. The width of the rim will depend upon whether the engine will use flywheel as a pulley or not, and upon the weight required. It should be remembered that the wider the rim the less will be its weight for a given amount of energy storage. The outside diameter is usually from four to five times the stroke of the engine. It is customary to limit the speed of the rim to 6,000 feet per minute, or a maximum diameter of $D = \frac{1910}{N}$[*]

wherein D = the outside diameter of the wheel and N = the r. p. m.

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[*] Kent's M. E. Pocketbook.

CHAPTER XXI.

BALANCE-WEIGHTS.

The proper method of balancing an engine, and also the correct formula for calculating the weight of the balance-weight, is a subject of much discussion among engineers. The writer finds it impossible to attempt to give what would be called average practice in this regard, so wide is the difference of opinion. From the conditions under which a gas engine operates, especially engines with a single cylinder, it is not possible to balance an engine *perfectly* by means of a rotating weight. If the engine be balanced for horizontal vibratory effect it is found that it is too heavily counterbalanced for vertical movement. It is therefore the custom to attempt a medium between the two. Some builders balance what they consider to be the rotating part of the engine, and make no allowance for the reciprocating parts; others balance the rotating weights, *and add* to this one half the reciprocating weight. By the first method only the crankpin, the crankarms and that portion of the connecting-rod which is considered as rotating, is balanced. This gives rise to two sets of formulas, which, strange to say, give apparently good results in each case. The question now remains, what proportion of the connecting-rod should be considered as having a rotating effect? This may best be answered by weighing the connecting-rod in the following manner: Support the piston end of the rod on a trestle or other convenient support, and let it rest at a point opposite the center of the bearing on a knife edge; support the crank end of the rod on a platform or other scales, also by a knife edge, and see that the center line of the rod is horizontal. The weight of the crank end as shown on the scale is that which should be taken for giving rotating effect. If, however, five eighths of the rod be taken as rotating, the result will not be far from wrong.

Taking the two cases cited above, the following formulas apply:

Let B = the weight of the balance-weight;
M = the weight of the crankpin + the rotating portion of connecting-rod;
K = weight of reciprocating parts, including the remaining portion

of the connecting-rod, the piston and the piston-rod and crosshead in engines that employ them;

J = weight of both crankarms;

m = crank-radius = one half the stroke of the engine;

j = the distance to the center of gravity of the crankarm from the center of the crankshaft;

q = distance to center of gravity of balance-weight from the center of crankshaft;

Then $B = \dfrac{Mm + Jj}{q}$ for balancing the rotating effect alone; (24)

And $B = \dfrac{(M + \frac{K}{2})m + Jj}{q}$ for balancing the rotating weight and one half the reciprocating weight. (24 a)

The proper place for the balance-weight is considered by a great many designers to be undoubtedly on the crankarms. There are many builders who place a counterweight on the flywheel near the rim, and some who core out the rim on the side of the flywheel nearest the crankpin. There are two advantages for the latter method; one is that the counterweight may be much lighter, and the other that it is cheaper to make. On the other hand, it is necessary

CHAPTER XXII.

FOUNDATIONS.

Without a good foundation, an engine may be expected to give more or less trouble by vibration, and in time work itself loose from such a foundation as has been provided for it. No engine should be bolted directly to a floor for anything other than a temporary job; and even when the engine is to be placed upon an upper floor a foundation should be built in a hanging frame below the floor. Foundations are usually built of either concrete, stone or brick. On top of the concrete is set a capstone, the best material for which is granite, limestone, bluestone, or any stone of a close-grained structure may be used for the purpose, but the ordinary sandstones will not answer.

Concrete should be made of good sand and a good quality of cement and broken stone. The stone should pass through a *two-inch ring* but not go through a one-to make some allowance in an increase of the size of the crankshaft over what would otherwise be necessary, in order that it will withstand the wrenching effect of the counterbalance when placed in the flywheel.

built upon it, should be placed as close to a wall as practicable. When building the foundation, it is necessary to insert in the concrete, gas pipe of an internal diameter equal to twice the outside diameter of the foundation bolts. The nut or threaded plate into which the foundation bolt screws is placed at the bottom of the tube, or the foundation bolt is set in with the tube. After the engine is in place the tubes should be filled with cement. Capstones are not necessary with the smaller engines, and it is often a good plan to lay wooden beams on top of the foundations, and to then put the engine on top of them, so that when the engine frame is bolted down it beds itself into the timber. The timber cap often stops an annoying vibration when it can be overcome in no other way.

In order to determine the proper dimensions of an engine foundation for any engine the following formula will be found useful:

Let $F =$ the weight of the foundation;
$E =$ the weight of the engine;
$R =$ the r. p. m.;
Then $F = .21\, E \sqrt{R}$ (25)

The weight of brick per cu. ft. is 112 lb. in the wall or foundation. The average weight of concrete is 137 lb. per cu. ft. It is not *customary* to make any difference in

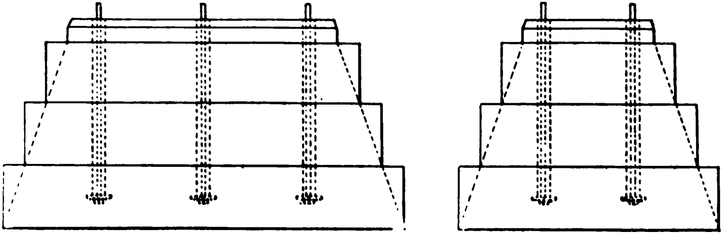


Fig. 38.

[illegible] foundation [illegible] dimensions [illegible] an average of the two [illegible]

A very good [illegible] trated in Fig. [illegible] of any material [illegible] The inclination of wall [illegible] bottom is usually shown [illegible] it is customary to make the [illegible] to 4" to the foot in height [illegible] building the foundation [illegible] some designers prefer to make [illegible] with a gradual slope, as [illegible] dotted lines. The number of [illegible] bolts varies from four to eight according to the size of the engine. No less than four bolts are used, and to determine roughly the necessary number of bolts, divide the horsepower of the engine by 6, always using an even number. The sizes of the bolts to use, may be determined approximately as follows:

Let H = the horsepower of the engine;
y = the area of one bolt;
k = the number of bolts;

Then $$y = \frac{.115\,H + 1.35}{k} \qquad (26)$$

It should be observed that this formula is entirely an empirical one, and that the designer should consider the strain upon the bolt that is farthest from a line directly under the crankshaft, in cases wherein the shaft is at an unusual distance from the top of the foundation. As a rule it will be found that the above formula gives bolts of ample size for engines of the usual design.

In case it is necessary to go very deep to find a solid bottom on which to set the foundation, piles may be driven over an area slightly larger than that taken up by the base of the regular foundation, or a crib of timbers laid in. A sub-foundation of timber or piles should be laid, if possible, at a sufficient depth to keep the timber covered with water or very damp earth.

CHAPTER XXIII.

MISCELLANEOUS FORMULAS.

The following formulas may prove themselves useful to those who design or handle gas engines:

For the diameter of the camshaft:

Let c = the diameter of the camshaft;
D = the diameter of the cylinder in inches;

Then $c = .057\,D + .625$.

For the volume of the muffler:

Let M = the volume of the muffler in cu. in.;
L = the length of the stroke in inches;

Then $M = 3.5\,D^2L$.

For a closed circulating system of cooling the cylinder the capacity of the tanks is made from 20 to 50 gallons per I. H. P. Thirty gallons is usually considered an ample allowance. The allowance for evaporation is 0.1 gallon per D. H. P. per hour. *For water flowing* through the jacket Mr.

Frederick Grover recommends 4½ gallons per I. H. P. per hour.

The proper size of marine engine to use for launches from 20 ft. to 50 ft. long may be roughly estimated by means of the following equation:

Let $B =$ the length of the boat on the load waterline in feet;
$H =$ the horsepower of the engine;

Then $H = \frac{B}{2} - 9$.

For the smaller craft, to which so many gasoline engines are applied, the usual rules for the diameter and pitch of a screw propeller do not seem to be applicable. The author has derived the following empirical formula for pleasure boats which use engines of from 1 to 12 horsepower.

Let $d =$ the diameter of the propeller in inches;
$H =$ the horsepower of the engine;
$R =$ the r. p. m.;

Then for a 3-bladed propeller with pitch and diameter equal,

$$d = \sqrt{\frac{20{,}500\,H}{R} + 164}$$

For a 2-bladed propeller the diameter should be about 8% greater than for the 3-bladed propeller. It may be a new proposition to many to make the pitch and the *diameter* equal, as it is the custom of a great many boat builders to make the pitch $1.3\,d$. One of the most successful launch builders in this country uses the ratio pitch = diameter, and speeds his engines up to correspond.

CHAPTER XXIV.

TESTING.

Every manufacturer of gas engines
a department known as the testing f
equipped more or less fully with appar
for determining, before the engine le
the factory, if it is working satisfacto
The primary objects of the test are, firs
find out if the governor is so set as to
the engine its proper speed; second
means of the indicator, to discover if
igniter is properly timed, whether or
the valves open at the proper points in
cycle, and to see if the compression
the right amount. The gas-engine in
tor diagram is of value for the determina
of other points in the working of the en
as has been discussed in Chapter XIV.
third object of the test is to determi
the engine is giving the required amou
horsepower at the pulley. This is k
as the delivered or brake horsepow
noted by either the letters D.H.P. or

It is determined by means of a form of absorption dynamometer known as the prony brake. The fourth object of the test is the determination of the fuel consumption of the engine per I. H. P. per hour.

The necessary apparatus for a gas-engine test, as it is usually carried out on the testing floor, is as follows: A good steam-engine indicator, with a piston having an area of ¼ square inch, and, if possible, a pencil movement of extra strength, is the principal requisite for determining the I. H. P.

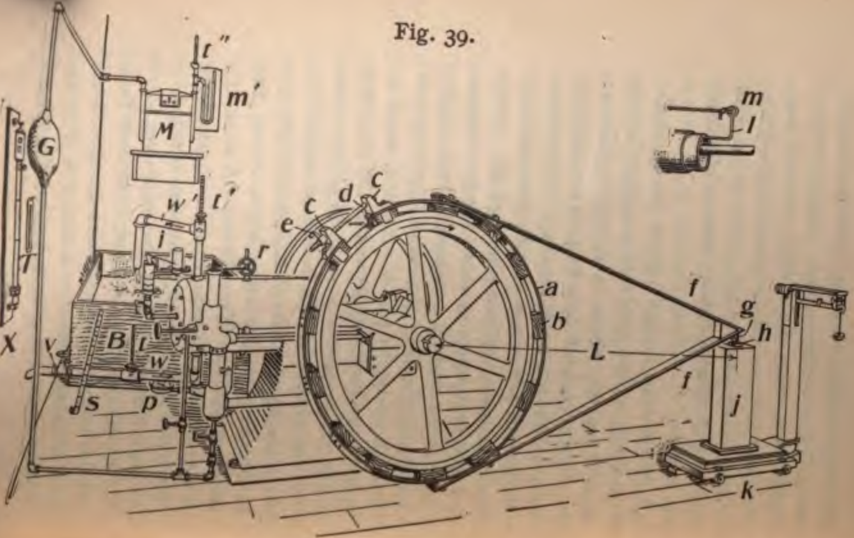
There are indicators now to be had that are designed expressly for the gas engine and with the above features. There should also be some good type of reducing motion in order that the movements of the stroke of the piston may be reduced to a stroke of between 2½″ and 3½″, which is the usual length of the indicator diagram. There should be, if it is possible to obtain it, a test meter reading in single cubic feet in order that the fuel consumption, when the gas engine is used, can be measured with some degree of accuracy. There should also be a prony brake large enough to be attached to a pulley of considerable diameter. The brake should have some means by which its grip upon the wheel may be quickly altered, and there should also be a platform scale, *and the* scale should be accurately tested beforehand with a standard weight to determine if its readings are correct. The experimenter should be supplied with a good form of revolution or speed counter, those being the best which have a soft rubber tip for placing in the center countersink at the end of the crankshaft.

For a more refined test than is usual on the testing floor of a factory there is needed a means of measuring the jacket water, thermometers for taking the temperature of the water before passing into the jacket and as it is leaving, a pyrometer for determining the temperature of the exhaust gases, a pressure gauge or a manometer for determining the pressure of the gas, a thermometer for determining the temperature of the same, a thermometer to be hung on the wall near the engine to determine the temperature of the room, and a barometer for measuring the pressure of the atmosphere at the time the test is made. This apparatus is necessary only if making the test for discovery of faults in design, and when it is desired to obtain data upon which to make improvements in the engine. It would, however, be a good plan to always take the pressure of the gas, particularly when natural gas is used and the pressure is more than a few ounces per square inch. Otherwise the experimenter may be led into grave errors by striving to calculate the fuel consumption of his engine. No gas engine should ever leave the factory without having undergone a test for speed and for delivered horsepower, nor without several indicator diagrams having been taken from the engine.

A gas engine rigged up for the purpose of making a thorough test is shown in Fig. 39. The prony brake is shown applied to the flywheel, and the engine is running over, as shown by the arrow. The brake consists of the strap *a* with the blocks *b* bearing upon the periphery of the wheel, and the brake-arm made of the two boards *f f*. Two cast-iron or wrought-iron angles *cc* are placed where the band is parted, and through them is put the bolt *d*, used to tighten the brake by means of the threaded crank *e*. A platform scale is placed at *k*, and the thrust of the brake is applied to the scale by the arm *f* acting through the knife edge *g*, the iron block *h* and the wooden stand *j*. The distance *L* from the center of the crankshaft to the point where the knife edge rests upon the block *h* is known as the length of the lever arm.

The indicator is shown at *i*, the cord from the indicator passing to the smaller drum of a reducing wheel *r*, and a cord from the larger wheel being attached to the piston.

Fig. 39.



as shown in the small sketch at the upper right-hand corner of the figure. To attach the cord to the piston a piece of 3/8″ round iron is bent into the shape illustrated and flattened at the end opposite the eye *m*. Through the flattened portion two holes are drilled, and the rod is fastened to the inside of the piston by means of machine screws or capbolts.

Water enters the water-jacket through the pipe *w*, controlled by the valve *v*, and it leaves the jacket through the pipe *w*. Thermometers are placed at *t* and *t* to determine the temperature of the water as it enters and as it leaves the jacket. These thermometers do not set directly in the water but are in small cups filled with oil.

In leaving the jacket the water is caught in the box *B* and either weighed or measured as is most convenient. In case it is desired to weigh the water, two receptacles should be provided and one emptied while the other is being filled. Catching the weight "on the fly" is not an accurate method and should be avoided. When using two receptacles, the stream of water should be changed just at the time the signal is given for taking the reading.

When it is desired to measure the water by volume, the receptacle may be made in the form of a box of the following dimen-

inch in depth will indicate
lb. of water. These measu
for water at a temperature
is about the average ten
water leaving the water-j
accurate determinations t
of course, be weighed. W
uring the water is necessa
a thorough test, in order to
of heat carried off in the w

A meter is shown on the
the gas bag at G. On the
bag, hangs the thermomete
ing the temperature of the
a barometer X for determi
pheric pressure. The tem
exhaust gases, as they leav
measured by means of t

fuel to the air under the various conditions. It has been the custom in many testing rooms to compute the volume of the air from the difference between the quantity of gas entering the cylinder at each working stroke and the volume of the piston displacement. Such a method is an erronous one, as there is usually more or less wire-drawing, and a certain quantity of the products of combustion remains from the previous charge, hence the air should be measured by a meter.

In order that the reader may thoroughly understand the method of making a complete test and of working up the data, it will be explained in its entirety. In ordinary testing he may select such portions of the complete test as will suit his purpose, and, before starting the test, a log should be made out as shown on page 188. This log shows at the head of each column just what readings should be taken. The last three columns are for convenience when working up the data after the test, while the balance of the log is for data taken while the test is in progress.

In making the test, particularly when a large number of readings are to be taken, the engineer in charge should have a sufficient number of assistants, in order that the readings may be taken immediately after

LOG OF TEST.

..................................*Gas Engine.* *Made by* ....

At .... 190...

				Temp., degrees Fah.				Wt.	Dynamometer.			Gas.		Air				
						Jacket Water.												
Number.	Time.	Rev. per minute.	Explosions per minute.	Room.	Exhaust.	Injection.	Discharge.	Pounds.	Scale reading, pounds.	Net pressure, pounds.	Foot-pounds per revol'n.	Cubic feet.	Pressure, inches of water.	Cubic feet.	Press. Barom.	M. E. P.	I. H. P.	D. H. P.
1																		
2																		

the signal. One man should be posted at the brake to keep the pressure upon the scale as nearly as possible at the same point during one run. Another man should have charge of the indicator, and a third should read temperatures, measure the water, and take the readings of the two meters and the barometer. This gives the third man the greatest part of the work, and a fourth man could therefore be used to advantage. The man in charge of the indicator may also take the speed of the engine and count the number of explosions per minute. For taking the speed of the engine, a continuous counter will be found more convenient than an ordinary speed counter, as the speed can be taken in less time. When there are but two observers, readings should be taken at intervals of ten minutes, and the man in charge of the indicator should take all readings, as the brake requires constant attention.

The test should be divided into runs, each run being made at the same horsepower throughout. One run should be made at the full power of the engine, another at the rated horsepower, and a third at no load with the brake off. The number of runs, at horsepowers other than the above, will depend upon the size of the engine and the time at the disposal of the experimenter. The author would suggest making a run at quarter load, half load, and three-quarter load as well as at the loads already noted. At least ten readings should be taken during each run, and fifteen or twenty readings would be much better, if accurate results are desired.

The test should, if possible, be in charge of a competent engineer. He should be provided with a whistle which should be blown as a preparatory signal thirty seconds before the time of taking the reading. Two blasts of the whistle should be the preparatory signal and one blast should be the signal for taking the readings. Promptness should be observed, each man taking his post at the preparatory signal. Under no circumstances should an outsider be allowed to interfere, and, above all, no one but an observer engaged upon the test should be permitted to take a reading. Always enter the data upon the regular log. Keeping notes upon loose slips of paper leads to confusion, and it should be avoided.

At some convenient time, either before or after the test, the diameter of the piston, the length of the stroke, and the clearance should be carefully measured. In order to measure the clearance, the crank should be placed exactly upon its inner dead center and the valves firmly seated. Carefully weigh a quantity of water and fill the compression space so that it is just full and no more, being careful to spill none of the water. Then weigh the remaining water and the difference between the two weights will be the quantity necessary to fill the compression space. Water at 39.1° F. weighs 62.5 lb. per cu. ft. and dividing the weight of the water taken to fill the compression space, by 62.5 gives the volume in cu. ft. If the water is much warmer than 39.1°, the weight may be found by the following formula:

$$\frac{62.5 \times 2}{\frac{t + 461}{500} + \frac{500}{t + 461}} = \text{Wt. per cu. ft.} \qquad (31$$

Wherein t = the temperature of the water.

This formula gives a close approximation to the correct weight. For the usual hydrant temperatures, dividing by 62.5 gives results sufficiently close for the purpose.

In making up a report of the test, the form shown on page 192 will be found convenient. The first line should be filled in with the name of the engine and the manufacturer, the second line with the name of the party by whom the test is made or the engineering firm of which he is the representative, and the third line by the locality at which the test is made, together with the date.

REPORT OF TEST.

.............................*Gas Engine.*

Test made by ....

At.................................*19*....

DIMENSIONS OF ENGINE.

Diameter of piston............	..In.	......
Area of piston................	Sq. in.	......
Length of stroke................	Ft.	......
Piston displacement	Cu. ft.	......
Clearance	Cu. ft.	......
Clearance..................	Per cent.	......

DATA.

Duration trial ....,....	Hrs.	......
Gas per hour.................	Cu. ft.	......
Air per hour	Cu. ft.	......
Ratio, gas to air..................		......
Jacket-water per hour............	Lb.	....
Jacket-water temperature, inlet......		......
Jacket-water temperature, outlet.....		......
Jacket-water temperature, range..	F.°	......
Revolutions per minute	Average.	.....
Revolutions per hour...............		......
Explosions per minute.....	Average.	......
Explosions per hour....		......
Temperature exhaust............	F.°	......
Temperature room.....	F.°	......

Length of lever arm..............Ft. | ......
Brake load, average..............Lb. | ......
Gas—Weight of cubic foot.......Lb. | ......
Air—Weight of cubic foot........Lb. | ......
Mixture—Weight of cubic foot...Lb. | ......
Specific heat, gas.................... | ......
Specific heat, air..................... | ......
Specific heat, mixture................ | ......
Heat value cu. ft. gas........B. T. U. | ......

RESULTS.

Work—Ft. lb. per min......Average. | ......
Work—Ft. lb. per hour.....Average. | ......
D. H. P.....................Average. | ......
Indicated M. E. P...........Average. | ......
Indicated H. P.............Average. | ......
Gas per I. H. P...............Cu. ft. | ......
Gas per D. H. P...............Cu. ft. | ......
Mech. Eff. D., H. P. ÷ I. H. P........ | ...
Friction Loss I. H. P.—D. H. P .... | .....

HEAT PER HOUR.

Supplied by gas..............B. T. U. | ......
Absorbed by jacket-water....B. T. U. | ......
Exhausted..................B. T. U. | ....
Absorbed in work...........B. T. U. | ...
Radiation...................B. T. U. | ......

Thermal efficiency...........Percent | ......
B. T. U. per I. H. P........ .... | ......

A great many of the items in the report need no explanation, and only those that are not likely to be understood by the reader will be explained. The percentage of clearance is found by dividing the volume of the clearance by the piston displacement. The ratio of the gas to the air is the quantity of gas used per hour divided by the quantity of air. In engines which take air into the cylinder at each cycle whether gas enters or not, the ratio may be obtained only when the engine runs without missing an explosion. The range of temperature for the jacket water is the difference between the temperatures of the outlet and the inlet water. The weight of the gas may be obtained from the gas company or it may be computed from the results of an analysis of the gas by multiplying the weight of a cubic foot of each constituent by the percent contained in the gas and adding the results. The following table gives the weights per cu. ft. of those gases which occur most frequently and in the greatest quantities in gases used for gas engines, together with their specific heat at constant volume.

	Lb. per Cu. Ft.	Spec. Heat
Marsh gas............	.0447	.470
Olefines..............	.1174	.332
Hydrogen	.00559	2.406
Carbon monoxide....	.0783	.173
Nitrogen	.0783	.173
Carbon dioxide......	.1060	.171
Oxygen	.1060	.155

These weights are for the gases when at an atmospheric pressure of 14.7 lb. per sq. in. and a temperature of 32° F. The specific heat of a mixture may be found in the same manner as the weight, by multiplying the specific heat for each gas by the percent contained in the fuel and adding the results. The weight of air at 32° F. and at a pressure of 14.7 lb. per sq. in. is .08082 lb. The specific heat of air at constant volume is .1688. The weight of a cu. ft. of the mixture may be found by multiplying the weight of the gas as already found by the percent of gas in the mixture, and then multiplying the weight of air per cu. ft. by the percent in the mixture. The sum of these two results will be the weight of the mixture. The heat value of the gas should in every case be determined at a laboratory. When it is necessary to know the heat value a sample of the gas should be sent to a laboratory for examination, taking samples of the gas at various times and mixing them to obtain an average. The heat values of various fuels will be found in Table I; but gas is an uncertain quantity in this respect, and the table is not to be relied upon when accurate results are desired.

Always, when making a gas-engine test, the volumes of the gases, both air and fuel, should be reduced to a standard temperature and atmospheric pressure for the purpose of comparison. It has long been the custom of engineers to use as standards the temperature of freezing water and the average pressure of the atmosphere at the sea level, or 30″ of mercury. To reduce the gas from the volume at which the test is made to its corresponding volume at 32° and at 30″ of mercury in the barometer, the following formula should be used:

Let T = the temperature at the time of the test;
I = the pressure in inches of mercury;
v = the volume at this pressure and temperature;
V = the volume at 30″ of mercury and at 32° F.

Then for air:

$$V = \frac{vI \times 493}{30 \times (T + 461)} \qquad (32)$$

This formula will also give a sufficiently close approximation when used for gas. The pressure by barometric measure has been used in this formula, as more convenient than the ordinary method of computing on the basis of pressure in lb. per sq. in. In order to reduce the pressure of the gas, when measured in inches of water, to an equivalent pressure in inches of mercury, divide the pressure in inches of water by 13.62. If the pressure of the gas is measured in lb. per sq. in., multiply this pressure by 2.033 to reduce to inches of mercury. These ratios are for temperatures of 32° F., and will be found sufficiently accurate for ordinary temperatures without temperature corrections.

The work in foot pounds per minute is computed from the indicator diagram. It is the continued product of the M. E. P., the area of the cylinder in sq. in., the length of the stroke in feet and the number of explosions per minute. The quantity of heat supplied by the gas per hour is the product of the heat value of the gas per cu. ft. by the number of cu. ft. used per hour. The heat absorbed by the jacket water is the product of the temperature range by the weight of the water supplied. The heat absorbed in work is the number of ft. lb. per hour divided by the number 778, as 778 ft. lb. is the equivalent of a B. T. U. (British thermal unit). A British thermal unit is the quantity of heat necessary to raise one pound of water through a temperature of one degree Fahrenheit. The heat carried off by the exhaust gases is found by the formula given below:

Let S = the specific heat of the mixture;
w = the weight of 1 cu. ft. of the mixture in pounds;
q = the quantity of the mixture in cu. ft. exhausted per hour;
T = the temperature of the room or the average temperature of the air and gas entering the engine;
T' = temperature of the exhaust as measured by the pyrometer;
U = the quantity of heat carried off by the exhaust in one hour.

Then $U = Swq\,(T' - T)$. (33)

The volume of the mixture passing through the exhaust, is the sum of the gas and the air used by the engine, and it includes all air passing through the exhaust for any reason.

The heat otherwise unaccounted for, and determined by subtracting the above three results from the heat supplied by the gas, is usually credited to loss by radiation. To be strictly accurate, the quantity of unburned fuel passing through the exhaust should be determined, and the heat value of this waste fuel subtracted from that supplied by the fuel taken into the engine.

The thermal efficiency is the quotient of the heat absorbed in work divided by the heat supplied to the engine in the fuel. It is usually written as a percentage, and it is the only basis upon which two engines should be compared for efficiency.

The indicated horsepower of a gas engine is determined from the indicator diagram. The area of the diagram should first be found by means of planimeter.*

The M. E. P. is determined by multiplying the mean ordinate by the scale of the indicator spring. In indicators using a special piston for indicating the gas engine, with an area of the spring of ¼ sq. in., the scale of the spring must be multiplied by two for the scale of the diagram, unless it is expressly marked for the smaller piston.

To find the I. H. P., when the M. E. P. is known, the following formula should be used:

---

*For the method of using the planimeter the reader is referred to works upon that subject, as space will not permit of its treatment in this volume. The area of the diagram should then be divided by its length, and the result will be the mean ordinate.

$$I.\,H.\,P. = \frac{Plan}{33{,}000} \qquad (34)$$

Wherein P = the M. E. P.;
l = the length of the stroke in feet;
a = the area of the piston in sq. in.;
n = the number of explosions per minute.

The reader should observe the difference between the value of n in this formula and value of the same letter in the similar formula for steam engines. In the formula for the steam engine, n is the symbol for the number of revolutions per minute.

To find the D. H. P. of the engine, the length of the brakearm—L, Fig. 39—the r. p. m. and the pressure exerted upon the scale must be known. The formula for the D. H. P. is:

$$D.\,H.\,P. = .0001904\,pln \qquad (35).$$

Wherein p = the pressure upon the scale (net);
l = the length of the brakearm;
n = the r. p. m.

The net pressure upon the scale should be found by subtracting the pressure exerted upon the scale by the unbalanced portion of the brakearm and the weight of the block j, Fig. 39, from the total pressure exerted upon the scale. In order to fin the effect of the unbalanced portion of the brakearm, proceed in the following manner: When the engine is not running, have a man stand upon the scale platform and balance his weight, *i. e.*, weigh him. Now balance the scale when he has hold of the brakearm and is pulling upward against a slight friction on the brakewheel or pulley, and again when the man is pushing down on the arm. The brakearm should be grasped close by the knife edge and the scale balanced when the brake is in motion. Add together the weight on the scale when the brakearm is being pulled up and the weight when it is being pushed down, and subtract the weight of the man from half this sum. The result will be the pressure due to the unbalanced portion of the arm. Thus, suppose the man to weigh 160 lb. and that when the arm is being pulled up the scale balances at 180 lb., while when the arm is being pushed down, the scale balances at 170 lb., the effect of the arm would be:

$$\frac{170 + 180}{2} - 160 = 15 \text{ lb.}$$

When making a brake test the man in charge of the brake should keep constant watch of the scale beam, with his hand always on the lever *e*, Fig. 39. Owing to the wide fluctuations in pressure upon the *piston of* a gas engine, especially in a hit-

and-miss engine, this is no easy task, and a good man should be selected for this post. The scale beam should be kept floating at all times, otherwise all the computations for D. H. P. will have no good foundation. When making a brake test of any duration, or when the engine is a large one, running water should be kept on the band while the engine is going.

On the factory testing floor, it is not customary to do any more than take the D. H. P., the I. H. P. and the fuel consumption. Sometimes a brake test only is made. It is always best, however, to take a sufficient number of indicator diagrams to determine if the ignition is properly timed and if there is any derangement of the valves.

CHAPTER XXV.

SELECTION.

In selecting a gas engine beware, first of all, of the oily tongue of the salesman. Every gas-engine manufacturer makes the "best gas engine on earth." Yet the market is overstocked with poor gas engines. To find out what gas engine is best adapted to your special line of work, consult several users of gas engines who are employing the engines for the same class of machinery which you intend to drive. The engine that will give satisfactory service pumping water may be a good engine for running an electric light plant, but it is doubtful if the regulation is sufficiently close for the purpose. On the other hand, the engine having heavy flywheels and a sensitive governing device, built expressly to meet the rigid requirements of an electric lighting service, is too costly a machine to be purchased for pumping water, because a cheaper engine *will answer* the purpose just as well, and the difference may be placed in the owner's pocket.

When you go to consult your neighbor with reference to his engine be prepared with questions regarding the operation of the engine, and also make it a point to see the engine in operation and watch it at work. Then find out as nearly as possible how much the engine costs the owner for repairs, and how often and for how long the engine has been laid up for repairs. Determine the time since the engine left the factory, and whether it has been in constant service ever since that time. If the engine is in a filthy condition, or if it is running with its parts out of joint when it is apparent that they could just as well be adjusted to run properly, score a few points in favor of the engine and against the engineer. An engine that will do fairly good work under bad management has something to recommend it. If, however, the engine is clean and all possible adjustments made, and yet is running in a noisy, jerky way, score a point against it.

If the engine is counterbalanced with the counterweights in the flywheel, stand in a position in line with the axis of the cylinder and observe if the flywheel is running true at the side. If it sways back and forth it is a sign that the crankshaft is not stra enough to withstand the strain produced by the counterweight. Determine if the bearings give much trouble from overheating, especially when the engine is working under full load. Ask if the igniter mechanism gives much trouble, and if frequent renewals of its various parts are necessary. In fact, the necessity for the frequent renewal of any part of the engine is a black mark against it.

Find out how much attention, on an average, the engine requires per day, and just what the nature of this attention is. If the engine is required to run a dynamo as a considerable proportion of its load, watch the lights when the dynamo is running. If the lights show no perceptible winking, when the light is observed indirectly, the engine will give satisfactory service for electric lighting purposes. A good way to test a light is to try to read fine print by it. If the light throbs to too great an extent, the throbbing will be quite distinct and the eyes will quickly tire. If the pressure on the lines is 100 volts, the swing of the volt-meter needle should not exceed two volts when swaying back and forth. On a 50-volt circuit this swing should not exceed one volt, or 2% of the pressure, in any case.

If the engine shakes considerably each

time it receives an impulse, the troub
due to an insufficient foundation, but i
shaking continues when the engine is
ning light, *i. e.*, between explosions
effect is due to improper balancing.

In general, select an engine with
fewest number of parts, and with the
so arranged that they may be easily rea
in case of accident or when repairs o
kind are necessary. Don't think, bec
the parts are encased, that the engin
simple one, for the opposite may be
case. An engine with an enclosed mec
ism may be a very good one for all
The point that the writer wishes to im
upon the reader is, that it should n
necessary to pull the entire engine to p
in order to make some insignificant r
or adjustment.

LENGTH OF A CYLINDRICAL TANK REQUIRED TO HOLD ONE GALLON.

Diam.		Length per gal. in feet.	Diam.		Length per gal. in feet.
Ft.	In.		Ft.	In.	
1		.1704	11		.00141
1	3	.108	11	3	.00134
1	6	.0756	11	6	.00129
1	9	.0556	11	9	.00123
2		.043	12		.00118
2	3	.0337	12	3	.00113
2	6	.0273	12	6	.00109
2	9	.0225	12	9	.00105
3		.0189	13		.00101
3	3	.0161	13	3	.000970
3	6	.0139	13	6	.000934
3	9	.0121	13	9	.000901
4		.0106	14		.000868
4	3	.00943	14	3	.000838
4	6	.00841	14	6	.000809
4	9	.00755	14	9	.000782
5		.00681	15		.000756
5	3	.00617	15	3	.000732
5	6	.00564	15	6	.000708
5	9	.00515	15	9	.000686
6		.00472	16		.000665
6	3	.00435	16	3	.000645
6	6	.00403	16	6	.000625
6	9	.00374	16	9	.000606
7		.00347	17		.000589
7	3	.00323	17	3	.000572
7	6	.00303	17	6	.000556
7	9	.00283	17	9	.000540
8		.00266	18		.000525
8	3	.00250	18	3	.000511
8	6	.00235	18	6	.000497
8	9	.00222	18	9	.000484
9		.00210	19		.000471
9	3	.00199	19	3	.000459
9	6	.00189	19	6	.000448
9	9	.00179	19	9	.000437
10		.00170	20		.000426
10	3	.00162	20	3	.000415
10	6	.00154	20	6	.000405
10	9	.00147	20	9	.000395

WROUGHT IRON WELDED STEAM, GAS AND WATER PIPE.

Nominal Internal.	DIAMETER.		Thickness.	TRANSVERSE AREAS.		Nominal Weight per foot.	Number of Threads per inch of Screw.	Tap Drill.
	Actual External.	Actual Internal.		External.	Internal.			
Inches.	Inches.	Inches.	Inches.	Sq. Ins.	Sq. Ins.	Pounds.		
1/8	.405	.27	.068	.129	.0573	.241	27	3/8
1/4	.54	.364	.088	.229	.1041	.42	18	15-32
3/8	.675	.494	.091	.358	.1917	.559	18	19-32
1/2	.84	.623	.109	.554	.3048	.837	14	3/4
3/4	1.05	.824	.113	.866	.5333	1.115	14	31-32
1	1.315	1.048	.134	1.358	.8626	1.668	11½	1 7-32
1¼	1.66	1.38	.14	2.164	1.496	2.244	11½	1 17-32
1½	1.9	1.611	.145	2.835	2.038	2.678	11½	1 25-32
2	2.375	2.067	.154	4.43	3.356	3.609	11½	2 7-32
2½	2.875	2.468	.204	6.492	4.784	5.739	8	2 11-16
3	3.5	3.067	.217	9.621	7.388	7.536	8	3 5-16
3½	4.	3.548	.226	12.566	9.887	9.001	8	3 13-16
4	4.5	4.026	.237	15.904	12.73	10.665	8	4 5-16
4½	5.	4.508	.246	19.635	15.961	12.34	8	4 13-16
5	5.563	5.045	.259	24.306	19.99	14.502	8	5 11-32
6	6.625	6.065	.28	34.472	28.888	18.762	8	6 11-32
7	7.625	7.023	.301	45.664	38.738	23.271	8	7 7-16
8	8.625	7.982	.322	58.426	50.04	28.177	8	8 7-16

CIRCUMFERENCES AND AREAS OF CIRCLES

ADVANCING BY EIGHTHS.

Diam.	Circum.	Area.	Diam.	Circum.	Area.
1-64	.04909	.00019	**1.**	3.1416	.7854
1-32	.09818	.00077	1-16	3.3379	.8866
3-64	.14726	.00173	⅛	3.5343	.9940
1-16	.19635	.00307	3-16	3.7306	1.1075
3-32	.29452	.00690	¼	3.9270	1.2272
⅛	.39270	.01227	5-16	4.1233	1.3530
5-32	.49087	.01917	⅜	4 3197	1.4849
3-16	.58905	.02761	7-16	4.5160	1.6230
7-32	.68722	.03758	½	4.7124	1.7671
			9-16	4.9087	1.9175
¼	.78540	.04909	⅝	5.1051	2.0739
9-32	.88357	.06213	11-16	5.3014	2.2365
5-16	.98175	.07670	¾	5.4978	2.4053
11-32	1.0799	.09281	13-16	5.6941	2.5802
⅜	1.1781	.11045	⅞	5.8905	2.7612
13-32	1.2763	.12962	15-16	6.0868	2.9483
7-16	1.3744	.15033			
15-32	1.4726	.17257	**2.**	6.2832	3.1416
			1-16	6.4795	3.3410
½	1.5708	.19635	⅛	6.6759	3.5466
17-32	1.6690	.22166	3-16	6.8722	3.7583
9-16	1.7671	.24850	¼	7.0686	3.9761
19-32	1.8653	.27688	5-16	7.2649	4.2000
⅝	1.9635	.30680	⅜	7.4613	4.4301
21-32	2.0617	.33824	7-16	7.6576	4.6664
11-16	2.1598	.37122	½	7.8540	4.9087
23-32	2.2580	.40574	9-16	8.0503	5.1572
			⅝	8.2467	5.4119
¾	2.3562	.44179	11-16	8.4430	5.6727
25-32	2.4544	.47937	¾	8.6394	5.9396
13-16	2.5525	.51849	13-16	8.8357	6.2126
27-32	2.6507	.55914	⅞	9.0321	6.4918
⅞	2.7489	.60132	15-16	9.2284	6.7771
29-32	2.8471	.64504			
15-16	2.9452	.69029	**3.**	9.4248	7.0686
31-32	3.0434	.73708	1-16	9.6211	7.3662

⅜	10.603	8.9462
7-16	10.799	9.2806
½	10.996	9.6211
9-16	11.192	9.9678
⅝	11.388	10.321
11-16	11.585	10.680
¾	11.781	11.045
13-16	11.977	11.416
⅞	12.174	11.793
15-16	12,370	12.177
4.	12.566	12.566
1-16	12.763	12.962
⅛	12.959	13.364
3-16	13.155	13.772
¼	13.352	14.186
5-16	13.548	14.607
⅜	13.744	15.033
7-16	13.941	15.466
½	14.137	15.904
9-16	14.334	16.349
⅝	14.530	16.800
11-16	14.726	17.257
¾	14.923	17.728
13-16	15.119	18.190
⅞	15.315	18.665
15-16	15.512	19.147
5.	15.708	19.635
1-16	15.904	[illegible]

6.
⅛
¼
⅜
½
⅝
¾
⅞

7.
⅛
¼
⅜
½
⅝
¾
⅞

8.
⅛
¼
⅜
½
⅝
¾
⅞

9.

CIRCUMFERENCES AND AREAS—*Continued.*

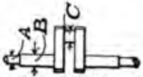
Diam.	Circum.	Area.	Diam.	Circum.	Area.
10. 3/8	32.594	84.541	**15.** 1/4	47.909	182.65
1/2	32.987	86.590	3/8	48.302	185.66
5/8	33.379	88.664	1/2	48.695	188.69
3/4	33.772	90.763	5/8	49.087	191.75
7/8	34.165	92.886	3/4	49.480	194.83
11.	34.558	95.033	7/8	49.873	197.93
1/8	34.950	97.205	**16.**	50.265	201.06
1/4	35.343	99.402	1/8	50.658	204.22
3/8	35.736	101.62	1/4	51.051	207.39
1/2	36.128	103.87	3/8	51.444	210.60
5/8	36.521	106.14	1/2	51.836	213.82
3/4	36.914	108.43	5/8	52.229	217.08
7/8	37.306	110.75	3/4	52.622	220.35
12.	37.699	113.10	7/8	53.014	223.65
1/8	38.092	115.47	**17.**	53.407	226.98
1/4	38.485	117.86	1/8	53.800	230.33
3/8	38.877	120.28	1/4	54.192	233.71
1/2	39.270	122.72	3/8	54.585	237.10
5/8	39.663	125.19	1/2	54.978	240.53
3/4	40.055	127.68	5/8	55.371	243.98
7/8	40.448	130.19	3/4	55.763	247.45
13.	40.841	132.73	7/8	56.156	250.95
1/8	41.233	135.30	**18.**	56.549	254.47
1/4	41.626	137.89	1/8	56.941	258.02
3/8	42.019	140.50	1/4	57.334	261.59
1/2	42.412	143.14	3/8	57.727	265.18
5/8	42.804	145.80	1/2	58.119	268.80
3/4	43.197	148.49	5/8	58.512	272.45
7/8	43.590	151.20	3/4	58.905	276.12
14.	43.982	153.94	7/8	59.298	279.81
1/8	44.375	156.70	**19.**	59.690	283.53
1/4	44.768	159.48	1/8	60.083	287.27
3/8	45.160	162.30	1/4	60.476	291.04
1/2	45 553	165.13	3/8	60.868	294.83
5/8	45.946	167.99	1/2	61.261	298.65
3/4	46.338	170.87	5/8	61.654	302.49
7/8	46.731	173.78	3/4	62.046	306.35
15.	47.124	176.71	7/8	62.439	310.24
1/8	47.517	179.67	**20.**	62.832	314.16

Diam.	Circum.	Area.	Diam.	Circum	Area.
20. ⅛	63 225	318.10	**25.**	78.540	490.87
¼	63.617	322.06	⅛	78.933	495.79
⅜	64.010	326.05	¼	79.325	500.74
½	64.403	330.06	⅜	79.718	505.71
⅝	64.795	334.10	½	80.111	510.71
¾	65.188	338.16	⅝	80.503	515.72
⅞	65.581	342.25	¾	80.896	520.77
21.	65.973	346.36	⅞	81.289	525.84
⅛	66 366	350.50	**26.**	81.681	530.93
¼	66.759	354.66	⅛	82.074	536.05
⅜	67.152	358.84	¼	82.467	541.19
½	67.544	363.05	⅜	82.860	546.35
⅝	67.937	367.28	½	83.252	551.55
¾	68.330	371.54	⅝	83.645	556.76
⅞	68.722	375.83	¾	84.038	562.00
22.	69.115	380.13	⅞	84.430	567.27
⅛	69.508	384.46	**27.**	84.823	572.56
¼	69.900	388.82	⅛	85.216	577.87
⅜	70.293	393.20	¼	85.608	583.21
½	70.686	397.61	⅜	86.001	588.57
⅝	71.079	402.04	½	86.394	593.96
¾	71.471	406.49	⅝	86.786	599.37
⅞	71.864	410.97	¾	87.179	604.81
23.	72.257	415.48	⅞	87.572	610.27
⅛	72.649	420.00	**28.**	87.965	615.75
¼	73.042	424.56	⅛	88.357	621.26
⅜	73.435	429.13	¼	88.750	626.80
½	73.827	433.74	⅜	89.143	632.36
⅝	74.220	438.36	½	89.535	637.94
¾	74.613	443.01	⅝	89.928	643.55
⅞	75.006	447.69	¾	90.321	649.18
24.	75.398	452.39	⅞	90.713	654 84
⅛	75.791	457.11	**29.**	91.106	660.52
¼	76.184	461 86	⅛	91.499	666.23
⅜	76.576	466.64	¼	91.892	671.96
½	76.969	471.44	⅜	92.284	677.71
⅝	77.362	476.26	½	92.677	683.49
¾	77.754	481.11	⅝	93.070	689.3
⅞	78.147	485.98	¾	93.462	695.
			⅞	93.855	70

Diam.	Circum.	Area.	Diam.	Circum.	Area.
30.	94.248	706.86	**35.**	109.956	962.11
⅛	94.640	712.76	⅛	110.348	969.00
¼	95.033	718.69	¼	110.741	975.91
⅜	95.426	724.64	⅜	111.134	982.84
½	95.819	730.62	½	111.527	989.80
⅝	96.211	736.62	⅝	111.919	996.78
¾	96.604	742.64	¾	112.312	1003.8
⅞	96.997	748.69	⅞	112.705	1010.8
31.	97.389	754.77	**36.**	113.097	1017.9
⅛	97.782	760.87	⅛	113.490	1025.0
¼	98.175	766.99	¼	113.883	1032.1
⅜	98.567	773.14	⅜	114.275	1039.2
½	98.960	779.31	½	114.668	1046.3
⅝	99.353	785.51	⅝	115.061	1053.5
¾	99.746	791.73	¾	115.454	1060.7
⅞	100.138	797.98	⅞	115.846	1068.0
32.	100.531	804.25	**37.**	116.239	1075.2
⅛	100.924	810.54	⅛	116.632	1082.5
¼	101.316	816.86	¼	117.024	1089.8
⅜	101.709	823.21	⅜	117.417	1097.1
½	102.102	829.58	½	117.810	1104.5
⅝	102.494	835.97	⅝	118.202	1111.8
¾	102.887	842.39	¾	118.596	1119.2
⅞	103.280	848.83	⅞	118.988	1126.7
33.	103.673	855.30	**38.**	119.381	1134.1
⅛	104.065	861.79	⅛	119.773	1141.0
¼	104.458	868.31	¼	120.166	1149.1
⅜	104.851	874.85	⅜	120.559	1156.6
½	105.243	881.41	½	120.951	1164.2
⅝	105.636	888.00	⅝	121.344	1171.7
¾	106.029	894.62	¾	121.737	1179.3
⅞	106.421	901.26	⅞	122.129	1186.9
34.	106.814	907.92	**39.**	122.522	1194.6
⅛	107.207	914.61	⅛	122.915	1202.3
¼	107.600	921.32	¼	123.308	1210.0
⅜	107.992	928.06	⅜	123.700	1217.7
½	108.385	934.82	½	124.093	1225.4
⅝	108.778	941.61	⅝	124.486	1233.2
¾	109.170	948.42	¾	124.878	1241.0
⅞	109.563	955.25	⅞	125.271	1248.8

Diam.	Circum.	Area.	Diam.	Circum.	Area.
40.	125.664	1256.6	**45.**	141.372	1590.4
⅛	126.056	1264.5	⅛	141.764	1599.3
¼	126.449	1272.4	¼	142.157	1608.2
⅜	126.842	1280.3	⅜	142.550	1617.0
½	127.235	1288.2	½	142.942	1626.0
⅝	127.627	1296.2	⅝	143.335	1634.9
¾	128.020	1304.2	¾	143.728	1643.9
⅞	128.413	1312.2	⅞	144.121	1652.9
41.	128.805	1320.3	**46.**	144.513	1661.9
⅛	129.198	1328.3	⅛	144.906	1670.9
¼	129.591	1336.4	¼	145.299	1680.0
⅜	129.983	1344.5	⅜	145.691	1689.1
½	130.376	1352.7	½	146.084	1698.2
⅝	130.769	1360.8	⅝	146.477	1707.4
¾	131.161	1369.0	¾	146.869	1716.5
⅞	131.554	1377.2	⅞	147.262	1725.7
42.	131.947	1385.4	**47.**	147.655	1734.9
⅛	132.340	1393.7	⅛	148.048	1744.2
¼	132.732	1402.0	¼	148.440	1753.5
⅜	133.125	1410.3	⅜	148.833	1762.7
½	133.518	1418.6	½	149.226	1772.1
⅝	133.910	1427.0	⅝	149.618	1781.4
¾	134.303	1435.4	¾	150.011	1790.8
⅞	134.696	1443.8	⅞	150.404	1800.1
43.	135.088	1452.2	**48.**	150.796	1809.6
⅛	135.481	1460.7	⅛	151.189	1819.0
¼	135.874	1469.1	¼	151.582	1828.5
⅜	136.267	1477.6	⅜	151.975	1837.9
½	136.659	1486.2	½	152.367	1847.5
⅝	137.052	1494.7	⅝	152.760	1857.0
¾	137.445	1503.3	¾	153.153	1866.5
⅞	137.837	1511.9	⅞	153.545	1876.1
44.	138.230	1520.5	**49.**	153.938	1885.7
⅛	138.623	1529.2	⅛	154.331	1895.4
¼	139.015	1537.9	¼	154.723	1905.0
⅜	139.408	1546.6	⅜	155.116	1914.7
½	139.801	1555.3	½	155.509	1924.
⅝	140.194	1564.0	⅝	155.902	1934
¾	140.586	1572.8	¾	156.294	194
⅞	140.979	1581.6	⅞	156.687	19

TWO-CYCLE ENGINE DIMENSIONS.

Horsepower.	Cyl. Diam.	Stroke.	Rev. per Min.	Cyl. Wall	Jacket Wall	Depth Jacket	 Crankshaft.			Propeller.			Piston.		
							A.	B.	C.	Diam. Shaft.	Pitch.	Diameter.	Diameter.	Length.	Diam. Pin.
1	3½	3½	500	⅜	¼	½	1 ⅛	1 ¼	1 ¼	¾	24	14	3 ½	5	¾
2	4½	4	450	½	5/16	7/16	1 ½	1 ⅝	1 ⅝	1	28	16	4 ½	6	⅞
3	5½	5	425	½	5/16	⅝	1 ¾	1 ⅞	1 ⅞	1 ¼	32	20	5 ½	7	1 ¼
4	5½	6	400	⅝	5/16	11/16	1 ⅞	2	2	1 ⅜	42	24	5 ½	8	1 ½
6	6½	6	375	⅝	5/16	¾	2	2 ¼	2 ¼	1 ½	48	28	6 ½	9	2
8	7½	7	350	¾	⅜	¾	2 ¼	2 ½	2 ½	1 ¾	48	30	7 ½	10	2 ⅜
10	8½	8	325	¾	⅜	¾	2 ¾	3	3	2	50	32	8 ½	12	2 ¼
16	9½	9	300	¾	⅜	¾	3	3 ¼	3 ¼	2 ⅜	50	36	9 ½	14	2 ¼

The clearance varies with different makes from 30% to 40% of the piston displacement.

TWO-CYCLE ENGINE DIMENSIONS.

Horsepower.	Pipes.		Gas.	Flywheel.				Conn. Rod.			Ports.			
	Inlet.	Exhaust.		Diam. Marine.	Face Marine.	Diam. Stat'y.	Face Stat'y.	Length.	Width Piston End.	Width Crank End.	Exhaust.	Inlet.	Width.	Drop.
1	1 ¼	1 ½	¼	14	2 ½	18	2 ½	7	1 ¾	2 ¼	7/16	⅜	3 ¼	⅛
2	1 ½	1 ¾	⅜	15	3	20	2 ½	8	2 ½	2 ½	7/16	⅜	3 ¼	3/16
3	1 ¾	2	½	18	4	25	3	10	2 ¾	3	½	7/16	3 ½	¼
4	2	2	¾	20	4 ¼	25	3	12	3	3 ½	⅝	½	4 ¼	¼
6	2 ¼	2 ½	¾	22	4 ½	30	3 ½	14	3 ½	4	9/16	⅝	4 ¾	¼
8	2 ½	3	1	22	4 ½	38	4	15	4 ¼	5	¾	⅝	5 ½	¼
10	3	3 ½	1	24	5	42	4	16	5	5 ½	¾	⅝	7	¼
16	3 ½	4	1	26	5	48	5	18	5 ½	6	⅞	¾	7	¼

APPENDIX.

TWO CYCLE ENGINES.

The accompanying table of two-cycle engine dimensions is, in the main, self-explanatory. A few remarks may help to avoid confusion on the part of the reader. The horsepowers in the first column are the rated horsepowers of the engines, and are less than the maximum which the engines are capable of giving. Where not otherwise stated, the dimensions given are the diameters of the various parts, and all dimensions are in inches. The figures under "Cyl. Wall" and under "Jacket Wall" indicate the thickness in each particular case. The "face" of the flywheel is the thickness of the rim in a direction parallel to the axis of the crankshaft. The widths of the connecting rod ends are the lengths also of the respective pins. Under the head of "ports," the first column gives the distance across the exhaust port opening measured along the direction of the piston travel. The *second column* gives a like dimension for the inlet port. Under "Width" is given what is, strictly speaking, the length of the port, and this dimension is that measured along the circumference of the inside of the cylinder. The "Drop" is the distance measured along the direction of the piston travel that the edge of the inlet port is beyond the edge of the exhaust port. The edges considered are those first crossed by the piston on its outward or downward stroke, and the direction is that toward the crankshaft. Thus, in a two horsepower engine, the exhaust port is uncovered for a distance of ⅜ inch before the inlet port begins to open.

In a two-cycle engine it is of the utmost importance that there should be no leakage from the crank-chamber, and in order to insure good results in this respect it is advisable to make the length of the crankshaft bearing from 2½ to 3 times the diameter of the shaft, the latter being the dimension most preferred.

On the piston it is customary to use three rings, two of which are placed at the working end and the other at the crank end (as in Fig. 33 at Z) for the pump cylinder. It is *especially* important that the piston should *be an* exceedingly good fit to the cylinder, *and it* should, therefore, have a perfect bear*ing*. A good plan is to make the piston light-driving fit and to scrape the high portions. After a good bearing is obtained the crankshaft and the connecting rod are put in place with the piston, and a belt from the line shaft is put on the flywheel. The engine is then run by this means for several days in order to wear it to place. Some manufacturers even go far as to grind the bore of the cylinder on a grinding machine. The piston rings should be at least ⅛ inch wider than the ports. The ports themselves are either cored in the cylinder or milled in, according to the fancy of the maker. Bars are sometimes, but not always, left across the ports. Very large engines often have two bars.

Annoyance is sometimes experienced with both two-cycle and four-cycle engines when they are operating with jacket water supplied under pressure and even under the ordinary head of the closed circulating system by water getting into the cylinder from the jacket. For this reason all water jackets should be tested under a water pressure of at least 60 lb. after the cylinder has been bored. If leaks into the cylinder are discovered, the jacket should be filled with a mixture of water and sal ammoniac in the proportion of about two ounces of sal ammoniac to one quart of water, and this solution should be left to stand in the jacket [illegible] cases. The solution [illegible] may be in the [illegible] many castings [illegible] process.

[illegible] been used with [illegible] in the [illegible] large, governing [illegible] so that [illegible] consistency of [illegible] same manner is [illegible]

[illegible] two-cycle [illegible] pressure is [illegible] possible to make [illegible] The draftsman should [illegible] crank-case [illegible] as possible, [illegible] smallest amount of clearance [illegible] judgment, he believes it safe [illegible] the crank-case side plates [illegible] attention in this regard, and [illegible] the interior should be made [illegible] parts as closely as it is [illegible] them.

[illegible] cover all that portion [illegible] exposed to the heat of the

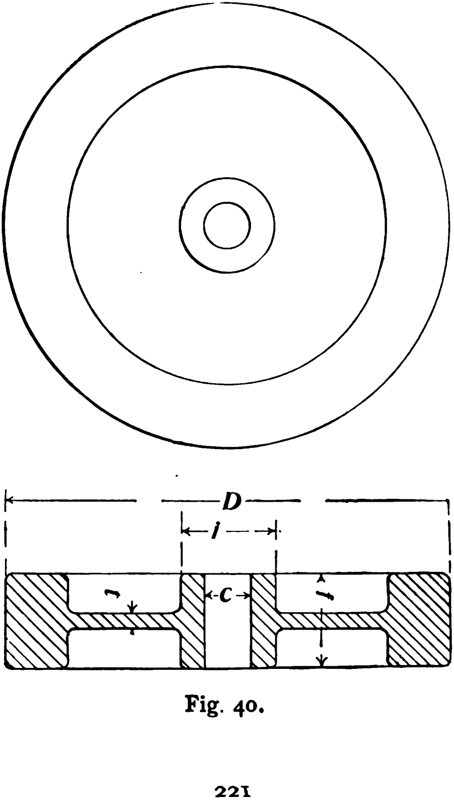


Fig. 40.

explosion, and the water should enter as close to the exhaust port as it is possible to place the inlet-pipe. It should leave the jacket at the highest point of the engine. The inlet should also be as close to the bottom of the jacket as possible. Avoid all pockets where steam can collect. A drain cock should be provided at the lowest point of the jacket in order that it may be emptied in freezing weather.

When designing the passages to and from the cylinder, make it a point to avoid square corners in order to prevent the formation of eddy currents in the gases. This is especially important regarding the passage from the crank-chamber to the cylinder, as eddy currents serve to choke the passages.

The proportions of a marine flywheel are shown in Fig. 40. The equations employed are made up from the average of two-cycle practice. The proportions are as follows:

$$f=2c$$
$$i=2c$$
$$t=\frac{c}{3}$$
$$D=3S$$

Wherein S=the stroke of the engine.

In the table of dimensions of the two-cycle *engines*, the last equation is not strictly *followed*, and the formula $D=2S+8''$ comes nearer to the average. It is not, of course, necessary to follow precisely the results given by either of these equations, but they give results that are very close to average practice. The thickness of the rim measured along the radius of the wheel depends upon the weight. This should be found by the use of formula (22) on page 163. When designing a marine flywheel it is necessary to make the diameter as small as practicable in order that crank-shaft may be low in the boat. It should also be remembered that the larger the diameter of the wheel the less will be its weight for the same service.

The length of the connecting rod, which is understood to be the distance between the centers of the bearings, is made twice the stroke by a great many builders, although some make it as short as 1½ times the stroke. By consulting the table it will be seen that the tendency is to make the stroke slightly less than the diameter of the cylinder. This is done in order to get the engine as short as practicable. While this gain is an advantage in marine engines, it is not an assistant to the fuel economy, but rather the reverse.

The drop of the inlet port below the exhaust is a much discussed question. In some of the earlier two-cycle engines the top edge of the inlet port was placed on a line with the lower edge of the exhaust port, thus giving a drop equal to the width of the exhaust port. In the table it will be seen that the drop is very slight, and it is claimed by some manufacturers that the drop should be zero, *i. e.*, the upper edges of the ports should be exactly in line. This is sometimes called "making the ports even."

It is now considered good practice to grind in the cylinder head, the igniter plug and other similar parts of the engine. Copper gaskets are also used for this purpose, especially in the joint of the igniter plug. Thick paper and sheet asbestos is also employed, while the side plates of the crank-case are often put up with a sheet-lead gasket.

On the air-inlet valve or other valves not subjected to high temperatures, leather is quite frequently used to stop the noise.

It is a good plan to pickle all castings, and a mixture of one part muriatic acid and twenty parts water will be found about right for this purpose. The castings should be left in the pickle for about 48 hours.

The use of brass bushings for vaporizer stems will prevent them from rusting. It is also well to use a brass bushing on all valve stems, especially if the engine is to be used in the neighborhood of salt water.

For the electrodes of a hammer-break igniter, brass pins have been found better than steel. Silver, or platinum, and, in many cases, a hard platinum-iridium alloy has been found to last better than either of the two first mentioned materials.

Candle wicking and ordinary brown soap have been found the best combination for packing gasoline valve stems, and soap should be used instead of oil or white lead for lubricating the threads of pipe on the gasoline circuit.

Both ends of the connecting-rod and the crankshaft bearings in the side plates should have brass bushings. The diameter of the piston pin is, approximately, ¼ that of the piston.

So design the engine that the piston and the valves may be removed in the shortest possible length of time and without special tools.

All cams, rollers and small pins, subject to wear, should be case-hardened when it is practicable to do so.

Many two-cycle engines having hammer-break igniters, have the igniter mechanism so arranged that the igniter will snap just before the end of the up-stroke, no matter which way the engine is running, and without changing the trip. On such engines it is *not* necessary to turn the engine over in the direction it is to run, called "cranking the engine." It suffices to swing the flywheel back and forth through a small arc until sufficient charge is pumped into the cylinder, and then to give the wheel a quick turn in a direction opposite to the running direction and far enough to secure an ignition, when the engine will start. If, when stopping, the igniter current is turned off before the gasoline valve or the throttle, sufficient mixture will remain in the cylinder to start the engine with one swing of the wheel. It will take a little practice to get the knack of this operation, but it is a time and a muscle saver.

A gentleman in Florida recently wrote the author that much annoyance would be saved the gas engine user if the cylinder oil were turned off a few minutes before stopping the engine previous to long shut down. This will allow the oil to be entirely burned out of the cylinder and save an accumulation of gum which will quite often bind the packing rings fast in their grooves.

. Splitting the packing rings of a gas engine piston by a straight cut, at an angle of about 45° to the side of ring, is not approved of by experienced designers. This method is good enough until the ring begins to wear, provided the two ends of the ring are close together at the start. As the rings wear, the split opens, and gases escape past the piston, causing loss of compression and a consequent loss of power. The better way is to divide the rings with a Z shaped split, the midline of the split being parallel to the sides of the ring. With a properly made division of this sort the ring will hold the gases back even when considerably worn.

JUMP-SPARK IGNITION.—For engines running at speeds higher than 800 r. p. m., the hammer-break igniter has been found to give considerable trouble. The jump-spark method is usually employed for speeds above 800 r. p. m. For a general description of this method the reader is referred to page 67. Instead of a ⅛-inch spark gap, as recommended on page 71, most makers of plugs now use a gap of from .02 to .04 inch, the latter being the thickness of a dime.

There seems to be quite a large difference of opinion among the makers of jump-spark coils as to just what combination produces the best results. For a coil that will operate successfully on a storage battery of one cell, an experienced electrical-instrument maker recommends the following: Make a core 5½ inches long of No. 22 iron stove-wire which has been carefully annealed, laying the wires side by side and using a *sufficient* number to make a bundle 1¼ inches in diameter. The wires should not be brightened, but should have a good covering of black oxide, with which they usually become coated during annealing. It will also improve the core, if the wires are dipped in thin shellac varnish and allowed to dry before being bunched. The core should then be bound together with a thread, in order to hold the wires together, and over it should be slipped a tube of fiber insulation, or it should be covered with two or three layers of paper that has been well soaked in paraffine wax. On this paper, or on the tube, should be wound three layers of No. 16 cotton-covered magnet wire, making the length of the winding about 5 inches long. This will require about 70 turns per layer and about 10½ ounces of wire. Next, cover this winding with paper or fiber insulation, and on this wind ½ lb. of No. 36 magnet wire. After the coil is finished it should be boiled in wax (½ beeswax and ½ paraffine wax), and placed in a wooden box which is large enough to leave a space of at least ⅜ inch between the box and the coil, and the space should be filled with wax. This coil will require two volts and about ¾ to 1 ampere of current to operate it.

Another combination recommended is a *core 1 inch* diameter, 6½ inches long, with a primary winding 6 inches long of three layers of No. 18 wire and a secondary of 2 lb. of No. 28. This coil will operate best on about 4 volts and 2 amperes. A battery of eight chloride-of-silver cells, connected four in series, has been used quite successfully in connection with a jump-spark coil.

A condenser in parallel with the primary coil, is employed to lessen the heat of the spark formed at the break of the primary interrupter. See Fig. 10 for method of connecting. For the first mentioned coil a condenser, having 2,400 sq. in. of surface on each side, or 4,800 sq. in. altogether, is recommended.

PREMATURE IGNITION.—In addition to what was said on this subject on page 37, it may be remarked that much of the trouble from premature ignition is caused by improper location or poor design of the igniter points. If these are long and slender they will become heated to incandescence after the engine has been running for a short time, and in some instances they have been known to burn off. Igniter points should be short and thick, and they should be so placed that they are in the path of the incoming new charge, in order to get the cooling effect of the charge. The plug should be surrounded with a water space as closely as practicable. Placing the igniter in the path of the exhaust gases, at the same time away from the coolin fluence of the fresh charge has been ductive of untold annoyance to at least prominent builder.

A sharp corner, especially if it is dir in the path of the exhaust, will freque collect carbon in thin flakes which become white hot after the engine has running a short while. While heatin air before it strikes the gasoline, is reso to in gasoline engine practice to assis the vaporization of the fuel; if it be ried to excess, premature ignition is li to result. For this reason it is advis to so arrange the heater that the tem ature may be at all times under the trol of the operator. This may be by having a by-pass around the heate that a portion of the air may pass dir to the intake, and then mix with the from the heater.

FOUNDATION BOLTS.—Instead of the and the threaded nut so generally emplo for the lower end of the foundation bol shown in Fig. 38, at least one gas-en maker has adopted the plan of turnin eye in the end of the bolt and slippi short piece of iron rod through it. T not only more convenient to make th threaded end and plate, but it effe prevents the bolt from turning when bolting down the engine.

BLAST FURNACE GAS ENGINES.—The difficulty of igniting the weak gases which are the by-product of the blast furnace has been entirely surmounted in its use as a fuel for gas engines. When compressed to pressures of from 110 to 135 lb. the gas ignites without difficulty. The expected difficulty with dust from the furnaces has been found to be more imaginary than real, and the elaborate cleaning devices which were at first employed are no longer used, the gas passing directly from the furnaces to the engine. In computing the dimensions of a gas engine for use on blast furnace gas, the value for X, in formula 5, on page 124, should be made 20,500, and the formula for blast furnace engines would then be

$$\text{D.H.P.} = \frac{D^2 \times L \times R}{20{,}500}$$

Many engines of the two-cycle type, with an air-pump in a separate cylinder, have been used for blast furnace gas. In these engines the air-pump first sends a charge of pure air into the working cylinder, clearing it entirely of all products of combustion (scavenging), and afterward the supply of gas and air is pumped in. The charge is, however, so controlled that none of it es-

capes through the exhaust port.
engines make X=10,300.

PRODUCER-GAS ENGINES.—Gas
operating on producer-gas made
thracite coal or from coke are bei
duced in locations where it is un
to use steam engines or gasoline
Gas engines will readily give an i
horsepower on a coal consumption
than 1.5 lb. per hour for the smal
and on less than one pound per
engines of several hundred hor
Engines operating on producer-gas
have a compression pressure of
lb., and with the former compres
give a mean effective pressure of
the maximum working capacity
would give a value of X in form
21,000 or not very much greater t
for the average engine.

Besides the ordinary producer-gas
employed for anthracite coal and co
is a combination of this with a w
generator, by means of which bit
coal may be employed for gas eng
manner similar to the aforem
fuels. Soft coal may be made int
the producer system, but the large
tion of tarry matter which is giv
in the process, can not be used
engine, and becomes a by-pro

combined water-producer-gas process overcomes this objection, and in some localities wood is also used as a fuel for this process. The author much regrets that he has been unable to obtain authentic data upon the dimensions required for a gas engine employing this gas.

AUTOMOBILE ENGINES.—In addition to the discussion in Chapter XIII, the following memoranda may prove of value. The gasoline tricycle is usually fitted with a 1¾ horsepower motor for ordinary grades, while for hill climbing 2¼ horsepower is generally employed. For single seated carriages for two persons six and eight horsepower seem to be the sizes of engines most in use. Drays are equipped with engines of twenty-five horsepower and upward. Engines having a water jacket are usually given a greater depth of jacket than is used for stationary engines. The author has noted, since the advent of the first edition, that several builders are using a curved shield for directing the air through the cooling tubes. The author does not, however, wish it to be inferred that this was necessarily the outcome of his suggestion.

The greatest trouble the automobile operator seems to have with a gasoline vehicle is with the ignition circuits. These are continually giving trouble, and, as a writer upon the subject has aptly remarked, "The troubles can nearly always be traced to poor electrical engineering." The author advises that, instead of using ordinary annunciator wire, as he has seen on several automobiles, at least No. 14 wire with the best weatherproof insulation be employed. The added cost is very small and the advantages gained in reducing the liability to breakages and bared insulation, will be worth many times the difference in cost. Every joint should be soldered when it is practical to do so, and all binding posts should be supplied with lock nuts, as the constant vibration of the vehicle has a tendency to loosen the connections.

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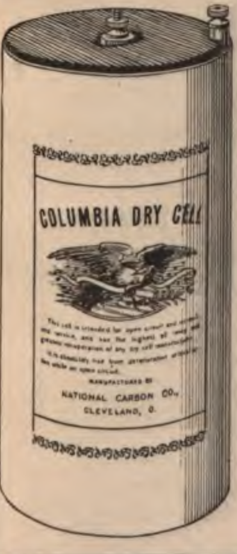

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